THE THEORY & DESIGN OF BRITISH SHIPBUILDING

By A. L. AYRE

HONOURS MEDALLIST AND KING'S PRIZEMAN

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The Theory and Design of British Shipbuilding.

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The Theory and Design of British Shipbuilding.

CHAPTER I.

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Dimensions. The measurements of ships are given in terms of **Length, Breadth** and **Depth.** While they are so often used by shipping people in stating the dimensions of a vessel, yet, owing to each of these terms being themselves measured in various ways, confusion is often caused and they are not always understood. It will, therefore, be advisable in this first chapter to explain the various points to which they are taken.

Length. The length most commonly used is *length between perpendiculars* (sometimes called builders' length). It is generally denoted by B.P., and is measured from the fore side of the stem to the after side of the stern post at the intersection of the line of the upper deck beams. (*See* Fig. 1).

The measurement of LENGTH for the purpose of determining scantlings is also measured in this manner according to the Rules of Lloyd's Register, except in the case of a cruiser stern where length of vessel is to be taken as 96 per cent. of the extreme length from fore part of stem in range of upper deck beams to the aftermost part of the cruiser stern, but it is not to be less than the length from forepart of the stem to after side of stern post, where fitted, or to the fore side of rudder stock, where a stern post is not fitted. British Corporation measure the length from the fore side of the stem to the aft side of the stern post, taken on the estimated summer load-line, where there is no rudder post, the length is to be measured to the centre of the rudder stock. *Registered length* is that measured

from the fore side of the stem to the after side of the stern post, as shown in Fig. 1.

Length overall is taken from the foremost part of the bowsprit or stem to the aftermost part of the counter.

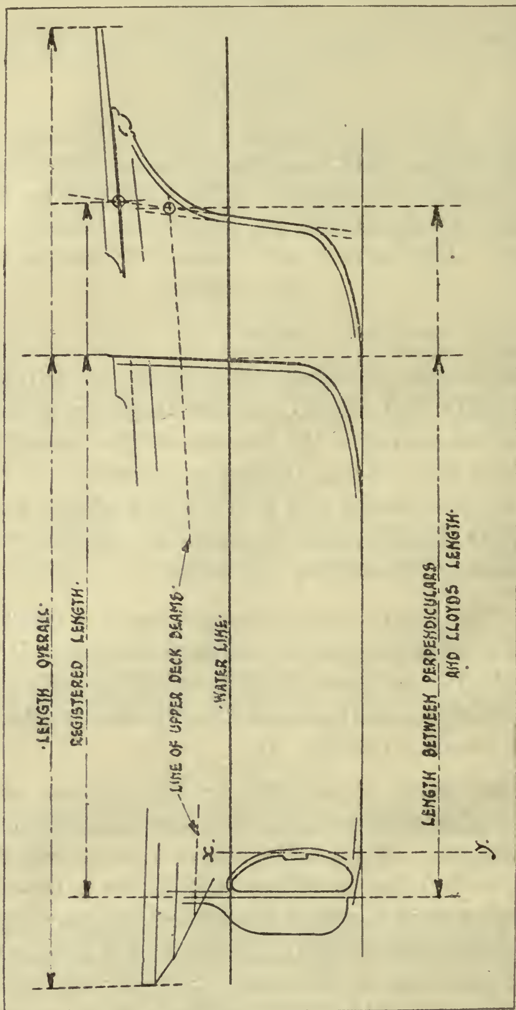
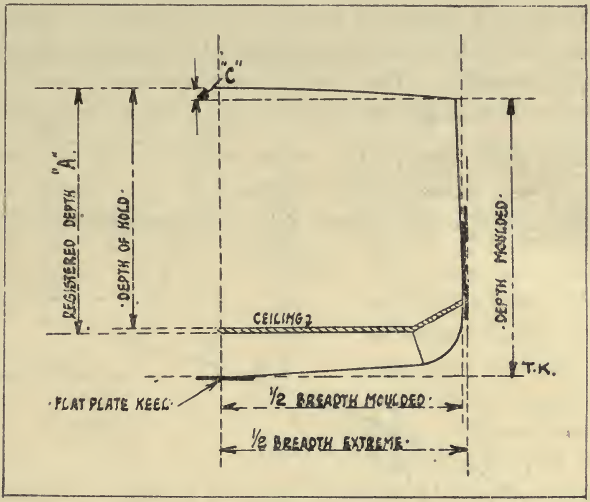


Fig. 1.

Breadth is usually given as *Moulded* or *Extreme,* and is measured at the widest part of the vessel. *Breadth Moulded* is measured over the frames, *i.e.*, from outside of frame on one side to the outside of frame on the opposite. Both Lloyd's and British Corporation

use the *Moulded Breadth* in finding the numerals for scantlings. *Breadth Extreme* is taken over the side plating at its widest part. (*See* Fig. 2). In the case of a vessel with fenders or sponsons another breadth would sometimes be given, as *Breadth over Fenders* or *Breadth over Sponsons*. *Registered Breadth* is exactly the same as *Breadth Extreme.*

Fig. 2.



Depth is measured at the middle of the vessel's length—*i.e.*, amidships, or *Centre between Perpendiculars* (**C.B.P.**). *Depth Moulded* is taken from the top of keel (**T.K.**) to the top of upper deck beam at the side of the vessel, at mid length between perpendiculars.

Lloyd's Register use the *Depth Moulded* in their numerals for scantling purposes.

British Corporation use the depth moulded, taken at the middle of the length of the load-line. *Depth of Hold* is measured from the top of ceiling (or tank top if ceiling is not fitted) to the top of beam at the centre line of the vessel and taken at 'midships. *Registered Depth*, measured at 'midships on the centre line of the vessel and to the top of beam, the lowermost extremity being the top of double-bottom, and if no double-bottom is fitted amidships, the top of floors. Should ceiling be fitted amidships, it is measured to the top, assuming its thickness to be 2½ in. in all cases. In Fig. 2, **A** shows the registered depth measured to the top of double-bottom.

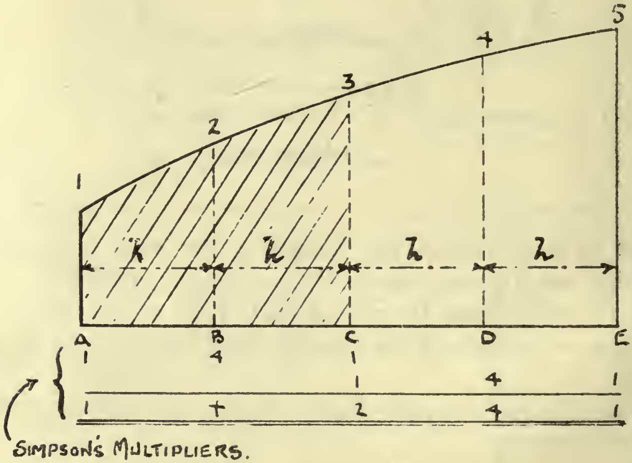
Camber of Beam is represented by **C** in Fig. 2.

Rules for finding Areas, Volumes, Moments and Centre of Gravity of Curvilineal Figures. A knowledge of these rules is a first necessity before engaging in ship calculations. This is very apparent, remembering the curved form of a vessel's hull.

Areas. The rules applicable to this work are : *Simpson's*, *Tchebycheff's* and the *Trapezoidal.* The second is, nowadays, coming more into vogue on account of its accuracy and quickness as compared with the others, particularly in the case of calculations referring to stability. The first is most commonly used in British yards, and will therefore be dealt with in the present chapter.

Simpson's First Rule. This rule assumes that the curve is a parabola of the second order. Suppose it is required to find the area of the portion of Fig. 3 from A to C, shown shaded. The portion

Fig. 3.



of the base A C is divided into two equal parts by the ordinate B, the common interval being h. Measure the lengths of the three ordinates A, B and C. To the sum of the end ordinates (of the shaded portion). A and C, add four times the middle ordinate B. The total so obtained and multiplied by one-third of the common interval h gives the area :

$$\tfrac{1}{3}h\ (A + 4\,B + C) = \text{area of figure from A to C.}$$

Now, suppose the total area of Fig. 3 is required—*i.e.*, from A to E—the interval h being equal throughout, the same rule could be applied to the three ordinates C, D and E, and the two areas so

found added together would give the total area from A to E. In Fig. 3 is shown the multipliers that would be used, 1, 4 and 1 from A to C, and 1, 4 and 1 from C to E. It will be seen that the ordinate C is twice taken into account; therefore, reading from A, the multipliers become 1, 4, 2, 4, 1. If the figure were further lengthened to the extent of two additional ordinates the multipliers would then be 1, 4, 2, 4, 2, 4, 1. It is obvious that this rule only applies to figures with an odd number of ordinates. If we number the ordinates from A, as is shown at the top of the sketch, we may put the rule into the following words: *Divide the base into an even number of parts and erect ordinates extending from the base up to the curve. To the sum of the lengths of the end ordinates add four times the length of the even numbered ordinates and twice the length of the intermediate odd numbered ordinates. The total so obtained multiplied by one-third of the commom interval gives the area.*

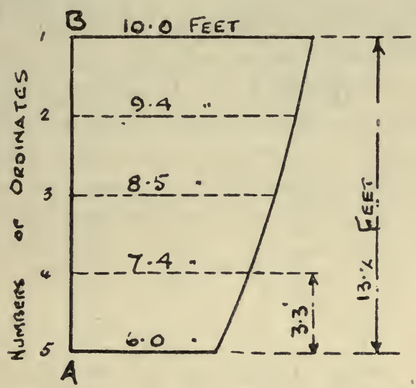


Fig. 4.

As an example, let Fig. 4 represent the section of a side coal bunker, the ordinates being as shown and the common interval 3·30 ft.

No. of Ordinate	Length of Ordinates	Simpson's Multipliers	Functions
1	10·0 feet.	1	10·0
2	9·4 ,,	4	37·6
3	8·5 ,,	2	17·0
4	7·4 ,,	4	29·6
5	6·0 ,,	1	6·0

Sum of functions $= 100{\cdot}2$

Multiplied by $\frac{1}{3}$ interval $= 3{\cdot}3 \div 3 = 1{\cdot}1$

Area $= 110{\cdot}22$ sq. ft.

Simpson's First Rule is the one most frequently used in ship calculations. However, in certain cases, where the positions of ordinates are given, this rule is not applicable; for instance, in the case of four, six, eight, ten, etc., ordinates. If four ordinates

are given, **Simpson's Second Rule** is employed. In this rule the multipliers are :

Four Ordinates = 1, 3, 3, 1
Seven ,, = 1, 3, 3, 2, 3, 3, 1
Ten ,, = 1, 3, 3, 2, 3, 3, 2, 3, 3, 1, etc.

The sum of functions obtained by using these multipliers is next multiplied by three-eights of the common interval, the result being equal to the total area. Example : Interval = 4 ft.

No. of Ordinate		Length of Ordinates		Simpson's Multipliers		Functions
1	...	10·0 feet.	...	1	...	10·0
2	...	9·4 ,,	...	3	...	28·2
3	...	8·5 ,,	...	3	...	25·5
4	...	7·4 ,,	...	2	...	14·8
5	...	6·0 ,,	...	3	...	18·0
6	...	3·9 ,,	...	3	...	11·7
7	...	1·0 ,,	...	1	...	1·0

Sum of functions = 109·2
Multiplied by $\frac{3}{8}$ interval = $\frac{3}{8} \times 4$ = 1·5

Area = 163·8 sq. ft.

Should an area be required between two consecutive ordinates, the **Five, plus Eight, minus One Rule** is used. Three ordinates are required. **Rule.**—*To eight times the length of the middle ordinate add five times the length of the other ordinate which bounds the required area and from this sum deduct the length of the acquired ordinate which lies outside the figure. The remainder multiplied by one-twelfth of the common interval will give the area.* Example : Suppose that in Fig. 4 the area between numbers 4 and 5 ordinates is required :

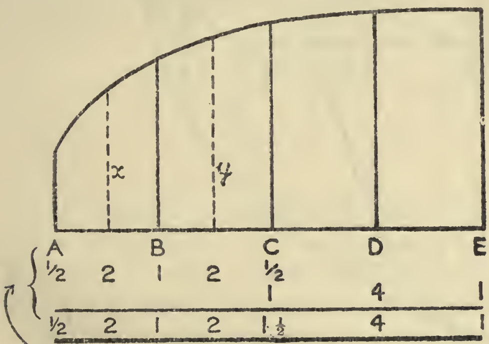
No. of Ordinate		Length of Ordinates		Multipliers		Functions
5	...	6·0 feet.	...	5	...	30·0
4	...	7·4 ,,	...	8	...	59·2
						89·2
3	...	8·5 ,,	...	1	...	8·5
						80·7

Multiply by $\frac{1}{12}$ interval = 3·3 × $\frac{1}{12}$ = ·275

Area = 22·19 sq. ft.

Six Ordinate Rule. If six ordinates are given, it will be seen that neither Simpson's First nor Second Rules can be used. By using the following multipliers : 1, 4, 2, $3\frac{3}{4}$, 3, $1\frac{1}{4}$, and the sum of functions multiplied by one-third of the commom interval, the area can be found. It is seldom, however, that six ordinates are used in any calculation, but when such does occur, the above rule is reliable.

Subdivision of Intervals is generally necessary in the case of the curve being of a very sharp nature at any particular point. For instance, in Fig. 5 the spacing of the ordinates between C and E would give a quite satisfactory result ; but between C and A, where there is a large amount of curvature, the result would not be so accurate, and it is advisable in such a case to subdivide the intervals as shown by x and y ordinates. The interval having been reduced by one-half, it is therefore, necessary to reduce the multipliers by the same amount. From A to C the multipliers will now become $\frac{1}{2}$, 2, 1, 2, $\frac{1}{2}$, and then commencing from C we have 1, 4, 1. Adding together the two multipliers for C, we have the new multipliers as shown in Fig. 5.



SIMPSON'S MULTIPLIERS

Fig. 5.

In applying these rules to the form of a ship, with the ordinates spaced longitudinally, it is usual to subdivide the end intervals because of the rounding in of the vessel's form at these parts. In Fig. 19 this will be seen in the sheer draught, where the first and last two intervals are subdivided.

Volumes. By means of the same rules we can find the volumes of curved bodies. Example : Let Fig. 6 represent a side coal bunker of a vessel. Divide its longitudinal length up into a suitable number of parts, say for Simpson's First Rule, as shown by the sections 1, 2, 3, 4, and 5. Find the area of each section by means of the rules, as was previously done in the case of Fig. 4, and then put these areas through the rule as follows :

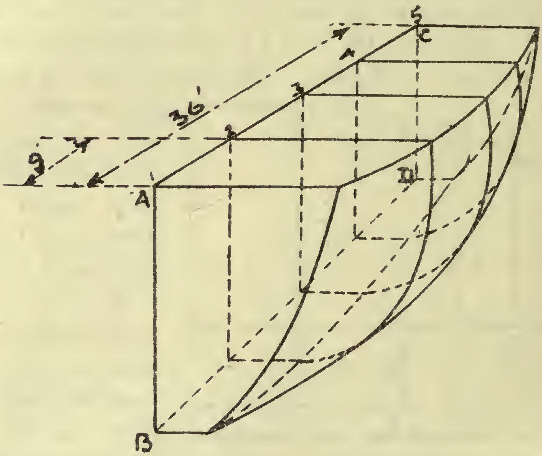
No. of Section	Area in Sq. Ft.	S.M.	Functions
1	89·54	1	89·54
2	95·25	4	381·00
3	90·03	2	180·06
4	81·25	4	325·00
5	60·44	1	60·44

Sum of functions 1,036·04

Multiply by $\frac{1}{3}$ of longitudinal interval $= 9 \times \frac{1}{3} =$ 3

Volume $=$ 3,108·12 cb. ft.

Fig. 6.



Having found the cubic capacity, the quantity in tons is easily found by dividing the total number of cubic feet by the number of cubic feet necessary to stow one ton of the coal that is to be put into the bunker. Taking the coal at 45 cubic ft. per ton as stowed in bunker, then 3,108·12 $\div$ 45 $=$ 69 tons that can be stowed in the above bunker.

Moments and Centres of Gravity. A Moment is a weight multiplied by a distance—for instance using the *ton* as the unit of weight and the *foot* as the unit of distance, we have the moment given in *foot-tons.* In Fig. 7 we have represented a bar 30 ft. long and a weight of 50 tons suspended from the point **D,** therefore 50 tons $\times$ 30 ft. $=$ 1,500 foot-tons *moment* acting about the point **A,** from which the distance is measured; or if the 50 tons were suspended at **C** we would have 50 tons $\times$ 20 ft. $=$ 1,000 foot-tons *moment.*

(These simple examples of moments are introduced here so that, in addition to the subject of Centre of Gravity, they may be of use in aiding the description necessary when considering the question of trim in later chapters, since the alteration of trim in any vessel is directly dependent upon the amount of **Moment** obtained by the weight multiplied by the distance it is moved in a fore and aft direction.)

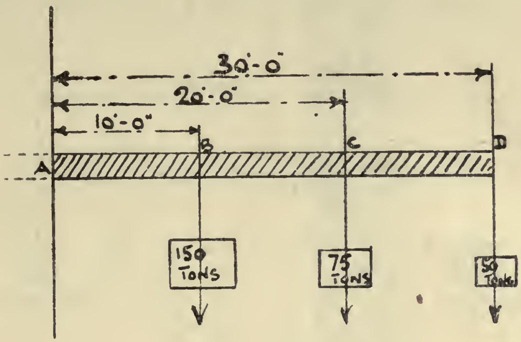


Fig. 7.

The distance which is generally termed *lever*, being reduced, has a very obvious effect in the reduction of *moment*. We also see that a small weight acting with a large leverage can have the came effect as a larger weight and small leverage. For instance, in this case, 50 tons at 30 ft. out, gives a moment of 1,500 foot-tons ; while 75 tons, at the point **C**, has a moment of exactly the same amount— 75 tons $\times$ 20 ft. $=$ 1,500 foot-tons moment.

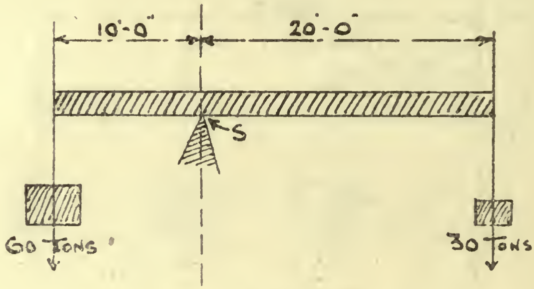
To find the amount of weight necessary to be placed at the point **B** so as to give the same amount as the 50 tons at **D** :

50 $\times$ 30 $=$ 1,500 foot-tons moment with 50 tons at **D** ; 1,500 foot-tons $\div$ 10 ft. (the distance out to **B**) $=$ 150 tons. Another way to look at the question is as follows : In Fig. 8 we have a bar supported at the point **S**, the overhang of one end being 10 ft. and 20 ft. the other, a weight of 60 tons being suspended from the short end. What weight would require to be hung on the opposite end to allow of the bar remaining horizontal ? Neglecting the weight of the bar, we have : Moment for short end $=$ 60 tons $\times$ 10 ft. $=$ 600 foot-tons, then 600 foot-tons $\div$ 20 ft. *lever* $=$ 30 tons required.

Centre of Gravity of Weights. *The Centre of gravity is the point about which we have the moments on either side balancing each other.*

It will therefore be obvious that, in the case of Fig. 8 the centre of gravity of the two weights lies upon the vertical line which passes through the point **S** since the moments on each side of that line are equal. In the case of a system of weights, the points at which we imagine the total to be concentrated so as to produce the same effect as the original distribution would be the centre of gravity.

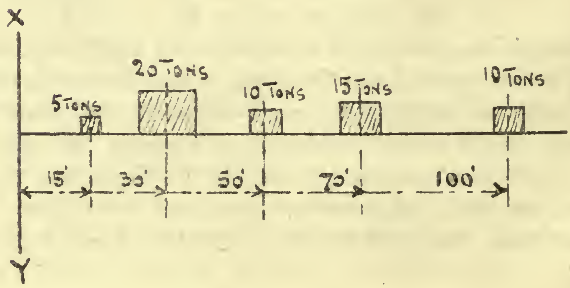
Fig. 8.



In Fig. 9 we have a system of weights, their quantities and disposition being shown. Find the centre of gravity relative to the vertical **XY** :

Weight		Distance from XY		Moment
5 Tons	×	15 Feet	=	75 Foot-tons
20 ,,	×	30 ,,	=	600 ,,
10 ,,	×	50 ,,	=	500 ,,
15 ,,	×	70 ,,	=	1,050 ,,
10 ,,	×	100 ,,	=	1,000 ,,
60 Tons Total Weight.				Total 3,225 Foot-tons.

Fig. 9.



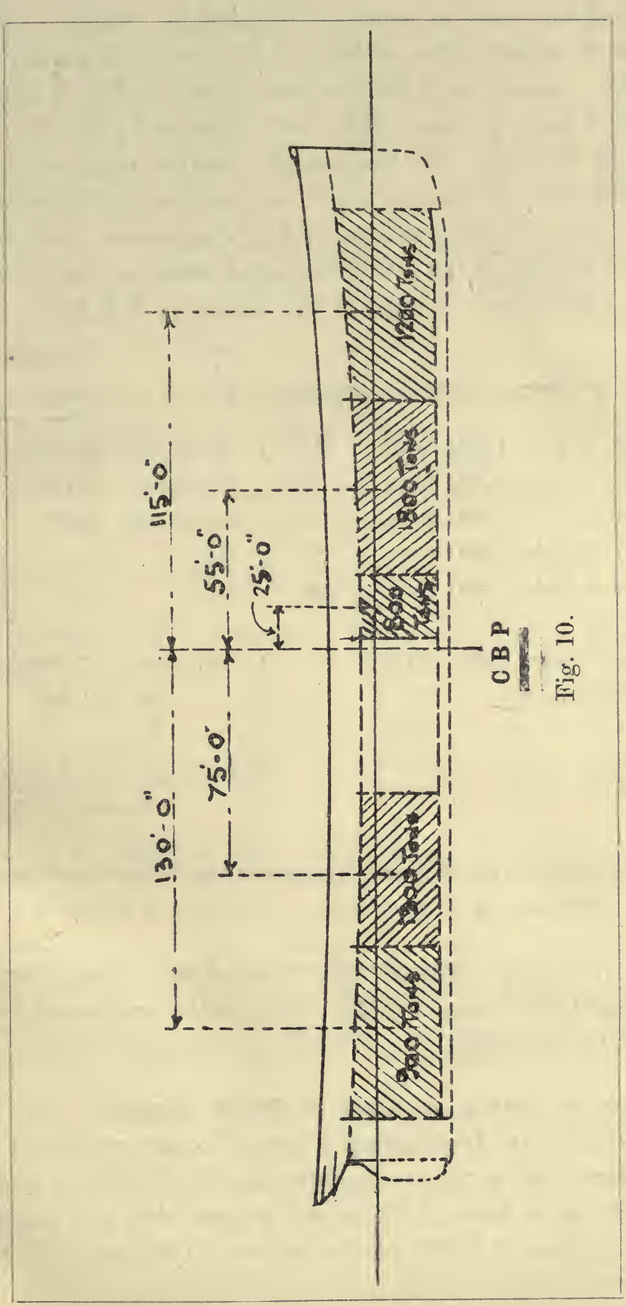


Fig. 10.

3,225 foot-tons total moment ÷ 60 tons total weight = 53·75 ft. from **X Y**, which is the position of the centre of gravity (**C G**), Therefore, we see that if 60 tons were placed at 53·75 ft. from **X Y**, we should have the same effect as is obtained by the distribution shown in the figure. To take a more practical example : Let Fig. 10 represent the profile of a vessel, her holds being filled with cargo of the amounts and positions of **C G**'s as shown, also bunkers. Find the position of the **C G** of cargo and bunkers, relative to 'midships.' ('Midships is denoted in the sketch by **C B P**) :

Position	Weight		Lever from C B P			Moments Forward	Moments Aft
No. 1 hold	1,200 tons	×	115	Ft.	(forward)	138,000	—
No. 2 hold	1,800 tons	×	55	,,	(forward	99,000	—
Bunkers	800 tons	×	25	,,	(forward)	20,000	—
No. 3 hold	1,300 tons	×	75	,,	(aft)	—	97,500
No. 4 hold	900 tons	×	130	,,	(aft)	—	117,000
Total Cargo and Bunkers	6,000				Forward	257,000	214,500
					Aft	214,500	
					Difference =	42,500 = net "Forward" Moment.	

The *forward* moment being in excess, the **C G** will obviously lie in that direction.

Net "forward" moment, 42,500 foot-tons ÷ total cargo and bunkers, 6,000 tons = 7·08 ft., which is the position of the **C G** forward of 'midships.

Centres of Gravity of Areas of Curved Surfaces. The Centre of Gravity of an Area having a curved boundary can be found at the same time as the area is determined by the use of Simpson's rules. If we go back to Fig. 4 and suppose that it is required to find the position of its **C G** relative to the end ordinate No. 1, proceed as follows :

No. of Ordinate	Length	S.M.	Functions	Levers*	Products for Moments
1	10·0	1	10·0	0	—
2	9·4	4	37·6	1	37·6
3	8·5	2	17·0	2	34·0
4	7·4	4	29·6	3	88·8
5	6·0	1	6·0	4	24·0
		Function of Area	100·2	Function of Moment	184·4

184·4 × common interval 3·3 ft. ÷ 100·2 = 6·07 ft. = distance of **C G** from No. 1 ordinate.

If the position of **C G** is required relative to the line **A B,** the method is as shown in the following, where the rule is: Half of of the sum of functions of "squares of ordinates" divided by the sum of functions for area equals the position of **C G** from the base line, from which the ordinates are measured.

In the case of the bunker (Fig. 4) we have:

No. of Ordinate	Length	S.M.	Functions	Square of Ordinates	S.M.	Function of Squares
1	10·0	1	10·0	100·00	1	100·00
2	9·4	4	37·6	88·36	4	353·44
3	8·5	2	17·0	72·25	2	144·50
4	7·4	4	29·6	54·76	4	219·04
5	6·0	1	6·0	36·00	1	36·00
Function of Area			= 100·2	Function for Moment		= 852·98

852·98 × $\frac{1}{2}$ ÷ 100·2 = 4·25 ft. = **C G** from **A B.**

* "Levers" represent the *number of intervals* from the given ordinate. The actual distance from the given ordinate could be used by multiplying the above *number of intervals* by their distance apart. The calculation is simplified by not making this multiplication until the sum of *products for moments* has been obtained, where the one multiplication serves the same purpose.

CHAPTER II.

Displacement Defined. Centre of Buoyancy Defined. Deadweight and Deadweight Scale. Composition of Deadweight. Tons per Inch. Difference in Draught.: Sea and River Water. The Block Co-efficient of Displacement. Co-efficient of Mid-Ship area. Prismatic Co-efficient. Co-efficient of Water Plane Area. Average Values of Co-efficients in Various Types. Relation of Co-efficients to Each Other.

Displacement. When a ship or any object is floating in water a certain amount of water is displaced from its position and put to one side. This amount of displaced water is obviously exactly equal in volume to the volume of the underwater portion of the vessel. If the amount of displaced water is measured and expressed either in volume or weight, we have what is known as *displacement.* However, if, instead of measuring the amount of water displaced, we measure the under water volume of the vessel to the outside of the skin plating, we have a practicable means of finding the amount of displacement. Suppose we have a graving dock filled with water up to the quay level and then a vessel to be lowered in from above. The result will, of course, be an overflowing of the water, the amount of which will be the *displacement* of the vessel—*i.e.*, the amount of water she has displaced. If the overflowing water were run off into a reservoir we could measure its quantity and express the amount either in volume or weight. The quantity of displaced water is dependent upon the weight of the vessel. *When a vessel is floating in equilibrium in still water she must displace an amount of water the weight of which is equal to the weight of the vessel.* We now, therefore, see that by *displacement* is meant the amount of water displaced (or put to one side) by a vessel, its amount being, in weight, exactly the same as the total weight of the vessel, consequently the word "displacement" is used to represent the total weight of a vessel. If we say that a certain ship is 5,000 tons displacement, we mean that her weight, including everything on board, is equal to that amount It is usually expressed in *tons*, being, first of all, measured in *cubic feet,* by calculations made from the drawing of the *lines* of the vessel's underwater form. Salt water being taken at 64 lbs. per cubic foot, we have $2{,}240 \div 64 = 35$ cubic feet of salt water per

ton, therefore, by dividing the amount of cubic feet (as found by the calculations) by 35, we have the vessel's displacement in tons. Fresh water weighing $62\frac{1}{4}$ lbs. per cubic foot, we have $2{,}240 \div 62\frac{1}{4} = 36$ cubic feet per ton.

Methods of Calculating Displacement are mentioned in a following chapter.

Centre of Buoyancy. *Centre of Buoyancy is the centre of gravity of the displacement of water—i.e., the centre of gravity of the volume occupied by the vessel's underwater portion*, or we may say: "It is the centre of gravity of the displaced water while in its original position—*i.e.*, before the introduction of the vessel." To determine the position of the Centre of Buoyancy of a ship, calculations are made, like the displacement, from the drawings of the *lines ;* of course, only dealing with the portion up to the water-line that the vessel is floating at. Its position is given in two directions—*fore and aft* and *vertically*. The fore and aft position, which is spoken of as the *Longitudinal Centre of Buoyancy*, is measured either from the *after perpendicular* or from *'midships ;* if the latter, the direction must be stated, *viz.*, *forward* or *aft* of 'midships. The vertical position, spoken of as the *Vertical Centre of Buoyancy*, is measured

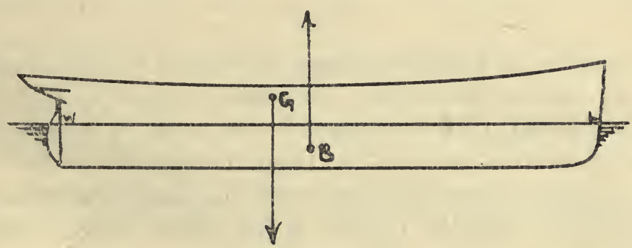


Fig. 11.

above the *top of keel*. In a following chapter the methods of obtaining the positions in these two directions are mentioned. The position of the **C B** must not be confused with the position of the vessel's **C G**. They are totally different, the Centre of Buoyancy being solely dependent upon the shape of the vessel's underwater portion, while the Centre of Gravity is determined according to the distribution of the vessel's structure and the various weights thereon. There is, however, a most important relation between the **C B** and **C G**. The **C B** being the centre of the supporting buoyancy, and the **C G** being the point through which the downward force of the

vessel's weight is acting, it will be obvious that for the vessel to be at rest in still water the two forces should act in the same vertical line—*i.e.*, the Centre of Buoyancy and the Centre of Gravity should lie on the same vertical line, thereby balancing each other. In Fig. 11 we have shown the longitudinal position of the Centres of Buoyancy and Gravity by **B** and **G** respectively, the direction of the two forces being also shown. It will be seen that this is an impossible condition for the vessel to be in, since the two forces are not acting through the same point. The vessel must, therefore, adjust herself until G and B are vertically in line, as is shown in Fig. 12. In Fig. 11 it is obvious that the tendency is for the vessel to trim by the stern. In Fig. 12 this condition is shown.

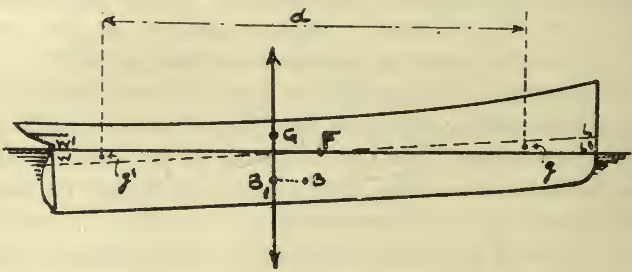


Fig. 12.

This alteration in trim has caused the Centre of Buoyancy **B** to shift aft, until it has become immediately below **G,** as is shown by $\mathbf{B}_1$. The shift of **B** is obtained by the wedge of buoyancy **W F** $\mathbf{W}_1$ being added by means of its immersion and the wedge **L F** $\mathbf{L}_1$ being deducted by means of its emersion. The displacement being equal in both cases, the volumes of the wedges must therefore be equal.

We now, therefore, see that *for a vessel to be floating in equilibrium in still water she must displace an amount of water, the weight of which is equal to the weight of the vessel, and also that the Centre of Gravity must lie in the same vertical line as the Centre of Buoyancy.*

Deadweight. This is the amount of weight that can be put on board a vessel in the shape of cargo, bunkers, stores, etc., after the completion of the structure and fittings and with the propelling machinery in steaming order. When completed in this condition, which is known as the **Light Draught,** the displacement is determined and spoken of as the **Lightweight.** All weight put on board

of a vessel in excess of the lightweight is **Deadweight.** If the lightweight of a vessel is 2,000 tons and then 4,500 tons of cargo, bunkers, etc., are put on board, the displacement will then become 6,500 tons in the load condition, this amount being the load displacement. We therefore see that the amount of deadweight a vessel is able to carry is the difference between the light and load displacements, or at any intermediate draught, it is the difference between the light displacement and the displacement at the particular draught. Fig. 13 represents a **Deadweight Scale** such as is generally supplied to a vessel by the builders. Upon it is shown a scale of draughts ranging between the light and load condition, opposite which is shown the corresponding amount of deadweight. In this case the light draught is shown to be 7 ft. 3 in., and since this corresponds to the lightweight, the deadweight will therefore be *nil.* The scale of deadweight is shown by intervals of 100 tons, until 4,500 tons is reached, at 19 ft. 6 in. draught, which is the maximum, and corresponding to Lloyd's Summer Freeboard. From this deadweight scale the amount of deadweight is readily ascertained for any particular draught within the range of the light and load. For instance, suppose it is required to know the amount of deadweight on board the vessel when floating at draughts of 15 ft. $10\frac{1}{2}$ in. forward and 17 ft. $1\frac{1}{2}$ in. aft :

$$\left.\begin{array}{l}15 \text{ ft. } 10\frac{1}{2} \text{ in.}\\ 17 \text{ ft. } \ \ 1\frac{1}{2} \text{ in.}\end{array}\right\} = \text{mean draught of 16 ft. 6 in.}$$

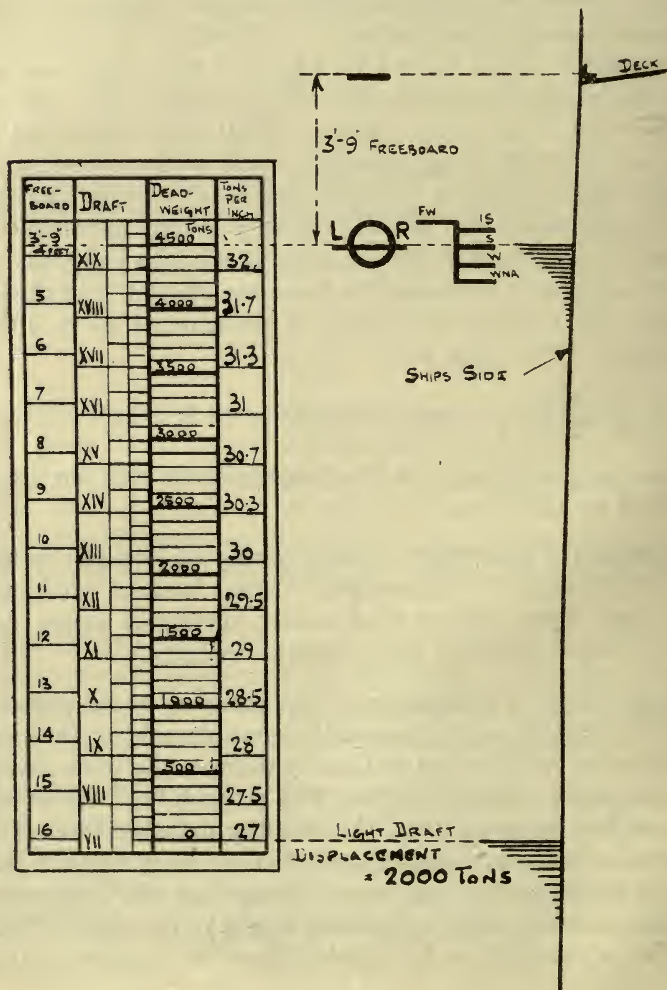
Upon the deadweight scale, at the draught of 16 ft. 6 in. the amount is read off as 3,350 tons.

Composition of Deadweight. The following are generally included under the heading of deadweight : Cargo, bunkers, consumable stores, fresh water, reserve feed water ; sometimes engine spare gear and water in donkey boiler are also included.

Tons per Inch. The number of tons necessary to be placed on board of a vessel so as to increase the mean draught to the extent of 1 in., or to be taken out to thereby decrease the mean draught by that amount, is known as the " tons per inch." If the draught is increased or decreased 1 in., the added or deducted layer of displacement is obviously equal in amount to the weight that is being added or deducted from the vessel, seeing that the displacement of the water must be equal to the total weight of the vessel. Therefore, if at any draught we obtain the amount of displacement con-

tained in a layer 1 in. thick, we will have the amount of weight corresponding to that layer. The tons per inch is found as follows: Assume that a waterline, at half-depth of a layer 1 in. thick, has an area which is a mean between the areas of the water-lines at the top and bottom of the layer. For instance, if we have a layer whose top water-line has an area of 6,000 sq. ft. and the bottom water-line 5,980 sq. ft. then the water-line at half-depth of layer is $\left.\begin{matrix}6{,}000\\5{,}980\end{matrix}\right\}$ 5,990 sq. ft., the mean between the top and bottom areas.

Fig. 13.



Now, by multiplying this mean water-line area by the depth of the layer, we obviously obtain the volume, which, if divided by 35 (salt water), will give the amount of displacement of the layer in tons, which, as stated above, is the " tons per inch." In the above-mentioned case we would have.

$$\frac{5{,}990 \text{ sq. ft.} \times \frac{1}{12} \text{ ft.} = \text{volume in cubic ft.}}{35}$$

$$= \frac{5990}{1} \times \frac{1}{12} \times \frac{1}{35} = \frac{5990}{420} = 14{\cdot}26 \text{ tons per inch.}$$

It will be seen that the "tons per inch" is equal to the area of the water-line divided by (12 × 35) or 420. In the case of a ship we have the areas of water-lines generally increasing as the draught deepens, therefore causing the tons per inch to alter. The number of tons necessary to increase or decrease the draught of a vessel to the extent of 1 in. from any particular draught being often required, it is, therefore, extremely useful to have the variation of tons per inch shown on the deadweight scale, as is shown in Fig. 13.

It may sometimes be necessary to approximate the tons per inch of a vessel when the calculated figures are not available. When such is required, the following formula may be used :

$$\text{Length} \times \text{Breadth} \div \text{c}$$

At the load draught :—	C	D
In modern full formed cargo steamer	470	70
In vessels of medium form	530	77
In fine lined vessels	600	88

Difference in Draught—Sea and River Water.

Salt water is taken at 64 lbs. per cubic ft. =

$$\frac{2240}{64} = 35 \text{ cubic ft. per ton.}$$

Fresh water is taken at $62\frac{1}{4}$ lbs. per cubic ft. =

$$\frac{2240}{62\frac{1}{4}} = 36 \text{ cubic ft. per ton.}$$

River water is usually taken at 63 lbs. per cubic ft. =

$$\frac{2240}{63} = 35 \cdot 555 \text{ cubic ft. per ton.}$$

When a vessel passes from salt to river water it is obvious that she must sink lower in the water, seeing that one ton of her weight will displace 35 cubic ft. of salt water, while, in river water, that amount of weight will displace 35·555 cubic ft. Therefore, for every ton of the vessel's weight we require ·555 cubic ft. extra volume of displacement when the vessel enters river water.

If **W** = the weight of the vessel in tons.

then **W** × ·555 = the addition, in cubic feet, to the volume of displacement when the vessel enters river water.

It is very often required to know what will be the *sinkage* of a vessel on her leaving salt water and entering river water. In Fig. 14 we have represented the section of a vessel both in salt and river

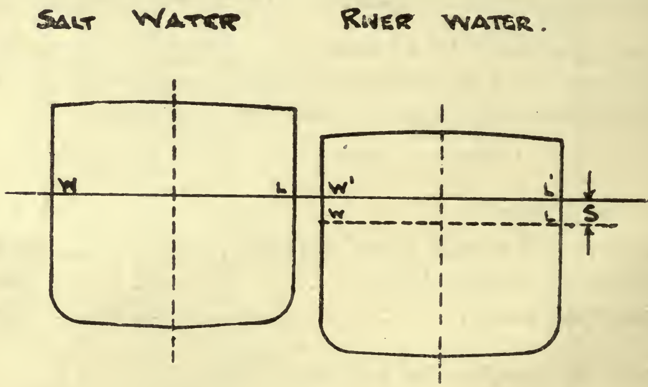


Fig. 14.

water. It will be seen that when in *river* the draught has been increased to the extent of **S,** the *sinkage.* **S** may also be termed the thickness of the layer, **W** $\mathbf{W_1}$ **L** $\mathbf{L_1}$. The layer containing the the required addition of volume in river water, it will be seen from the above, has a volume in cubic feet of :

$$\mathbf{W} \times \cdot 555.$$

Having found the volume of the layer, we can now proceed to find its depth, which is the amount of *sinkage :*

$$\text{Volume of layer} = \mathbf{W} \times \cdot 555 \text{ cubic ft.,}$$

$$\text{therefore its displacement in salt water} = \frac{\mathbf{W} \times \cdot 555}{35} \text{ tons.}$$

The displacement in tons of the layer, divided by the *tons per inch*, will obviously give the thickness of the layer in inches. Let $\mathbf{t}$ = the *tons per inch* in salt water at the water-line **W L** .

(The displacement of the layer having been taken in salt water, the *tons per inch* must therefore be taken in salt water. Of course, the displacement of the layer is really river water; it could have been taken as such, but, in that case, it would have been necessary to use a river water *tons per inch*. It not being usual to calculate the *tons per inch* for river water, we have taken the displacement in salt water, and by using a salt water *tons per inch* we obtain the same result as if we had dealt in river water.)

Displacement of layer $\div$ $\mathbf{t}$ = sinkage in inches.

$$\frac{\mathbf{W} \times \overset{1}{\cancel{555}}}{\underset{63}{\cancel{35}}} \div t = \text{sinkage in inches.}$$

$$\frac{\mathbf{W}}{63\mathbf{t}} = \begin{array}{c}\textbf{sinkage in inches when passing} \\ \textbf{from sea to river water.}\end{array}$$

While this simple formula is very useful, yet a shipmaster or officer may be debarred from its use owing to their not being in possession of the value of **W**, the displacement. The following is therefore given so that approximations may be made when only the form of the ship is known :

Draught $\div$ d = sinkage or rise in feet.

Values of d are given in above table.

The same formula is used when passing from river to sea water. Knowing the vessel's displacement and the *tons per inch*, it will therefore be a very simple thing for a ship's officer or any person to estimate the amount of *sinkage* or *rise*, as the case may be. For instance, suppose a certain vessel has to be loaded so that her draught on entering sea water will be 19 ft. The vessel loading in river, it will therefore be necessary to load her a little further than the 19 ft. on account of her rising on entering the sea water.

The amount of extra immersion while in the river would be as follows :

From the scale ascertain the amount of *displacement* in tons (*i.e.*, the *deadweight* plus the *lightweight*), also the *tons per inch.*

Referring to the scale in Fig. 13, we have at 19 ft. draught (salt water—

	4,310	tons	deadweight.
plus	2,000	,,	lightweight.
=	6,310	,,	displacement.

tons per inch = 32.

Then, by using the formula, we have :

$$\frac{W}{63t} = \frac{6{,}310}{63 \times 32} = 3{\cdot}13$$ in. of *rise* on the vessel entering salt water, therefore she may be loaded in the river down to a draught of 19 ft. 3 in., say.

Co-efficients. The Co-efficient most commonly used is the **Block Co-efficient** of displacement. *This is the ratio that the actual displacement of a vessel bears to the displacement of a block which has the same length, breadth and draught as the vessel.*

For instance, take a vessel 300 ft. long × 40 ft. breadth × 17 ft. 6 in draught. The displacement of a block having the same dimensions would be :

300 ft. × 40 ft. × 17·5 ft. = 210,000 cubic ft.
or 210,000 ÷ 35 (salt water) = 6,000 tons.

But suppose that, owing to the fineness of the ends, bilge, etc., the vessel has a displacement of only 4,500 tons.

The ratio that the vessel's actual displacement bears to the displacement of the block is $\frac{4{,}500}{6{,}000} = {\cdot}75$, which is the Co-efficent, or, written in the usual way :

$$\frac{4{,}500 \times 35}{300 \text{ ft.} \times 40 \text{ ft.} \times 17{\cdot}5 \text{ ft.}} = {\cdot}75.$$

Should the Co-efficient be known and the displacement be required :

$$\frac{300 \text{ ft.} \times 40 \text{ ft.} \times 17{\cdot}5 \text{ ft.}}{35} \times {\cdot}75 = 4{,}500 \text{ tons.}$$

It will be seen from the above that, by knowing a vessel's Block Co-efficient, we are in a position to form an idea of the shape of her underwater lines, to the extent of being able to say whether she is of *full*, *fine*, or *medium* form. For instance, a vessel whose Block Co-efficient is ·8 would be termed *full*, while ·5 would be *fine*, and ·65 *medium*.

'Midship Section Area Co-efficient. *This is the ratio that the immersed area of a vessel's 'midship section at any draught bears to area of a rectangle whose breadth is equal to the breadth of the vessel and its depth equal to the given draught.* Fig. 15 represents the 'midship section of a vessel floating at the water-line W L. The draught of the vessel is 17 ft. 6 in. T K, and the breadth taken over the widest part of the frame is 40 ft. The area of the circumscribing rectangle is therefore, 40 × 17·5 = 700 sq. ft.

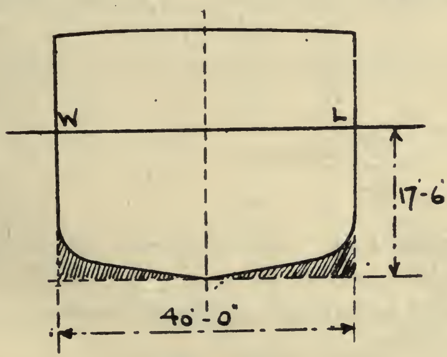


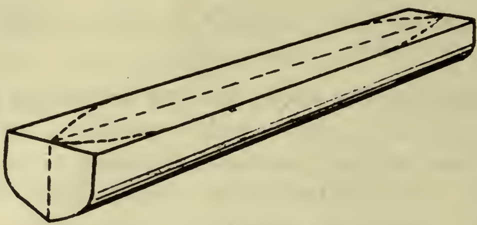
Fig. 15.

Let the actual immersed area of the vessel's 'midship section be 665 sq. ft., then $\frac{665}{700} = {\cdot}95$, which is the 'Midship Section Area Co-efficient. Should the Co-efficient be known and the *area* be required, 40 × 17·6 × ·95 = 665 sq. ft.

Prismatic Co-efficient of Displacement. *This is the ratio that the actual displacement of a vessel bears to the displacement of a prism whose length is equal to that of the vessel and the section of the same shape as the 'midship section of the vessel.* Fig. 16 shows such a

prism. This Co-efficient is found in a similar manner as the Block Co-efficient.

Fig. 16.



$$\text{Displacement of prism in tons} = \frac{\text{area of mid. sect.} \times \text{length}}{35 \text{ (salt water).}}$$

Using the above figures we have $\dfrac{665 \times 300}{35} = 5{,}700$ tons, and taking the actual displacement of the vessel at 4,500 tons, we have $\dfrac{4{,}500}{5{,}700} = \cdot789$ Prismatic Co-efficient, or, written in the usual way,

$$\frac{4{,}500 \times 35}{300 \times 665} = \cdot789.$$

Co-efficient of Waterplane Area. *This is the ratio that the actual area of a vessel's waterplane bears to the area of a rectangle whose length and breadth are equal to the length and breadth measured at the widest portion of the waterplane.*

Let Fig. 17 represent a waterplane of the vessel referred to above, the actual area being 10,800 sq. ft. The area of the circumscribing rectangle being 300 × 40 = 12,000 sq. ft., the Co-efficient will therefore be $\dfrac{10{,}800}{12{,}000} = \cdot9.$

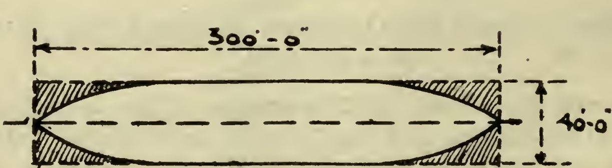


Fig. 17.

The dimensions used in all cases are *moulded—i.e., length between perpendiculars.** *breadth moulded,* and the draught taken from the *top of keel.* The displacement used is also *moulded—i.e.,* the displacements of the shell plating, bar-keel, or any other appendages are not included.

TABLE OF CO-EFFICIENTS GIVING THE AVERAGE FOR VARIOUS TYPES.

Type.	Block	Prismatic	'Mid. Set.	Waterplane	Approximate Dimensions Length	Breadth	Draught T.K.
Fast Atlantic Liner	·64	·67	·95	·75	650	70	30
Cargo and Passenger, about 16 kts.	·72	·75	·96	·82	520	61	27
Fast Cross-Channel	·50	·54	·92	·71	290	36	15
Large Cargo, about 12 knots	·77	·79	·98	·84	480	58	30
Medium Cargo, about 10 knots	·80	·81	·985	·87	350	48	24
Small Cargo, about 9 kts.	·77	·79	·98	·85	250	37	17½
Coaster, about 8½ knots	·73	·77	·95	·83	150	25	12
Screw Tug	·52	·59	·88	·73	130	27	10
Trawler	·48	·68	·71	·70	115	22	11
Herring Drifter	·43	·61	·70	·64	80	18	7
Sailing Vessel	·70	·74	·95	·80	265	42	20
Steam Yacht, about 14 knots	·44	·62	·71	·68	200	24	11
Steam Yacht, above 11 knots	·46	·63	·73	·70	125	20	9
Battleship	·63	·67	·94	·74	410	77	26½
Cruiser	·50	·55	·90	·63	500	71	26
Torpedo Boat Destroyer	·45	·64	·70	·66	210	21	8

Relation of Co-efficients to Each Order. The Block, Prismatic and 'Midship Section Area Co-efficients are closely related to each other, as is shown in the following, where it will be seen that, if any two Co-efficients be known, the remaining one can be found :

Block Co-eff. = Prismatic Co-eff. × Mid. Sec. Area Co-eff.
Pris. ,, = Block ,, ÷ ,, ,, ,,
Mid. Sec.
Area Co-eff. = ,, ,, ÷ Prismatic Co-efficient.

* In single-screw ships, where we have a propeller aperture, the length for displacement is sometimes measured to the foreside of the aperture, as shown by $x\ y$ in Fig. 1. When the displacement length is taken to the *perpendicular*, in such ships it is very fairly assumed that the displacement of the propeller and rudder counterbalances the effect of the aperture. Also, by this method, the displacement of the hull above the aperture is taken into account. In twin-screw ships the length taken is always the *length between perpendiculars.* For Co-efficient purposes, however, the *length between perpendiculars* is used in any case.

CHAPTER III.

ESTIMATE OF REQUIRED AMOUNT OF DISPLACEMENT. DETERMINATION OF DIMENSIONS. REQUIRED LONGITUDINAL CENTRE OF BUOYANCY AND TRIM. THE SHEER DRAUGHT. CONSTRUCTION OF "LINES" TO FULFIL THE REQUIRED CONDITIONS. FINAL "DISPLACEMENT SHEET" CALCULATIONS, INCLUDING LONGITUDINAL AND VERTICAL CENTRES OF BUOYANCY, ETC. THE DISPLACEMENT SCALE AND THE VARIOUS CURVES.

In the designing of a vessel the first step taken is to estimate the **required Amount of Displacement in the Load Condition.** The difference between the deadweight and the load displacement being the light displacement or *lightweight* of the vessel, we therefore see that, having given the deadweight, it remains for us to estimate the vessel's lightweight, the sum of which two will then give the load displacement required. First of all, an approximate estimate is made for the proposed dimensions, and with these and a corresponding approximate estimate of displacement the block co-efficient is determined. This preliminary determination is merely to ascertain if the resultant block co-efficient is one which is suitable for the vessel's required speed. It is fairly obvious that a fast vessel requires a finer co-efficient than a slow vessel. In high speed ships, where the wave-making resistance forms a large proportion of the total, it is imperative that the ends should be finely shaped so as to minimise the wave-making. The following formula is most useful in obtaining a suitable block co-efficient for ordinary vessels with a given speed :

$$1{\cdot}06 - \frac{\text{speed in knots}}{2 \times \sqrt{\text{length}}}$$

Should the preliminary co-efficient appear to be too full, the dimensions must necessarily be increased, or, if it is on the fine side, we can afford to decrease them.

We now have a basis to work upon, from which the actual design can be determined. These *proposed* dimensions must now be considered from an economical point of view. In the various laws and rules which apply to ships, grades occur where changes in the stipulations take place. These grades being dependent upon the vessel's dimensions, it is therefore necessary to ascertain in what position the proposed dimensions will place the vessel. Should it happen that she is just over a grade, it will generally be an advantage from an economical point of view, to make slight reductions in the proposed dimensions. (There are rare cases, however, where even a lighter and cheaper vessel can be built through the means of making the vessel on the high side of a grade.) By means of careful work, in this way it is possible to obtain a design in which the *cost per ton of deadweight* comes out at a low figure compared to one where the dimensions are fixed at random. Such a carefully designed vessel is also one in which the cost of maintainence and working expenses are low. Therefore, an owner, or some person on his behalf, thoroughly versed in all the intricate branches of ship design work, should carefully make all the necessary delicate investigations before making the final decision—a decision upon which depends not only the first cost, but the all-important paying capabilities of the vessel. Not only should the proposed vessel be considered from the *cutting down* point of view, but also from that of ascertaining what will be the resulting effect if additions are made. For instance, the author has known cases where, if a few feet had been added to the length of the vessel's bridgehouse, poop or forecastle, the draught would have been increased to the extent of 1 in. to 2 in., giving an addition of about 50 tons of deadweight at the expense of about £25, or the exceedingly low rate of about 10s. per ton for the additional amount of deadweight. This great advantage would have been due to a change taking place in the freeboard rules, the addition in length of the erections enabling the vessel to obtain the benefit of another rule. There are many things to take note of at this stage. For example, the *length*. In the calculation for tonnage the number of intervals taken, is governed by the vessel's length at various grades. It may be possible, by means of a slight deduction, to reduce the number of intervals by two, which would, in most cases, tend to reduce the vessel's tonnage. This reduction may not appear to be much, but still it has effect throughout the vessel's existence, and in the shape of working expenses amounts in the end, or per

annum, to a fairly large sum in favour of the owner. Again, perhaps it would be easily possible to avoid some heavy additional scantling required by the rules of registry to which the vessel is built. In the case of the *breadth*, we have it governing the number of rows of hold pillars and the size of the beams, therefore it should again be ascertained that we are not *just over a grade.* Depth also has similar effect. While depth may be termed the cheapest dimension, yet it is one in which great care is necessary in its decision. Depth very largely decides scantlings, particularly in the case of framing, while its relation to the length of ship is an important factor in determining the scantlings of longitudinal material in the top sides. Again, in the freeboard rules and tables we find spots where, by careful *juggling,* large advantages can be obtained. Therefore, by careful consideration with regard to the various stipulations contained in the laws for tonnage and freeboard and classification societies' rules, we are able by judicious handling to obtain, from an owner's point of view, a vessel which will be economical in initial cost, maintenance and working. In the determination of such dimensions, however, we must not lose sight of other important questions, such as speed and stability. In addition to the necessary consideration of fineness of the vessel's form, as previously mentioned, the proportions of the dimensions to each other should be suitable for the speed, and also from the important point of view of stability. Having now obtained suitable dimensions, the next proceeding is to estimate the lightweight. From the approximate estimate, the power necessary to drive the vessel at the given speed can be calculated, and again from this the weight of the machinery. Calculations are then made for the weight of the hull—iron and steel, timber and outfit—the sum of which added to the weight of machinery gives the lightweight. Next, the load draught is found by means of first estimating the freeboard according to the Board of Trade rules and tables. (The subject of Freeboard is dealt with in a later chapter.) This is a most intricate calculation, and there are many points to be watched in dealing with the various types of vessels if an accurate result is required. It is very important that such a result should be obtained, because in economical designing it is most desirable to know the exact draught that the vessel will be able to load down to, instead of having to allow a margin by means of a few inches in the vessel's depth. The freeboard being ascertained, the load draught is found as in the following example :—

	ft.	in.
Depth, Moulded	23	0
Depth of Keel	0	$1\frac{1}{2}$
Thickness of Deck Stringer Plate	0	$0\frac{1}{2}$
Statutory Deck Line above Stringer Plate...	0	2
Extreme Side	23	4
Certificate Freeboard	3	$2\frac{1}{2}$
Load Draught =	20	$1\frac{1}{2}$ to bottom of Keel.

The above particulars are shown in Fig. 18.

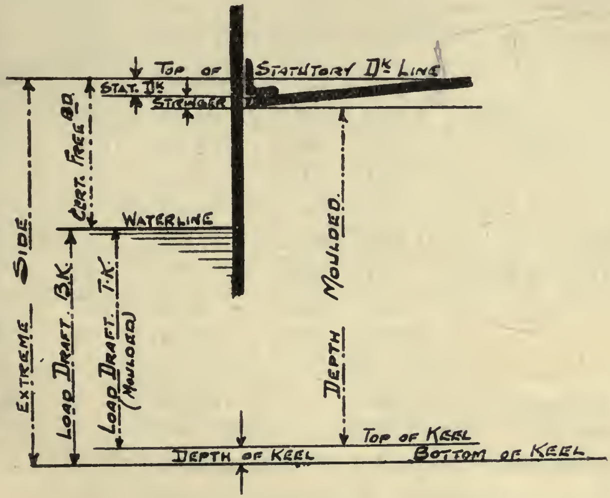


Fig. 18.

We now have the dimensions, lightweight, given deadweight and the load draught. Lightweight *plus* deadweight gives the load displacement. With this load displacement, length, breadth and draught we now obtain the co-efficient resulting from this preliminary estimate. It may be here necessary to again slightly modify the dimensions if the co-efficient is not near enough to that suitable for the vessel; however, at this stage we are able to finally decide the dimensions of the proposed vessel. Having arrived at the required dimensions, displacement and load draught, the drawing of the lines, commonly known as the sheer draught, can be proceeded with. If the *trim* of the vessel is specified, care must be

taken in fixing the position of the centre of buoyancy—as explained in Chapter II., where we saw that, for the vessel to be floating freely and at rest, it is necessary to have the centre of gravity of the vessel's weight, and the centre of buoyancy in the same vertical line, therefore at the desired trim we must have this occurring. We must first of all estimate the position of the centre of gravity, and then the position of the centre of buoyancy can be fixed to give the required trim. By means of an ordinary calculation of *moments* the position of the centre of gravity can be found, as shown in the following :—

Item.	Weight in Tons.	*Vertical Lever above Keel in Feet.	*Vertical Moment	Horizontal Lever from Aft Perpendicular, in Feet	Horizontal Moment
Hull, Iron and Steel	1,320	15·5	20,460	158	208,560
Wood and Outfit ...	350	26·0	9,100	170	59,500
Machinery	320	11·0	3,520	138	44,160
Lightweight	1,990	16·62	33,080	156·88	312,220
Stores and Fresh Water	25	23·0	575	130	3,250
Bunkers	315	21·0	6,615	157	49,455
Cargo	4,300	17·5	75,250	168	722,400
Load Displacement ...	6,630		115,520		1,087,325

* Vertical centre of gravity in load condition :

$$\frac{115{,}520}{6{,}630} = 17{\cdot}42 \text{ ft. above keel.}$$

Longitudinal centre of gravity in load condition :

$$\frac{1{,}087{,}325}{6{,}630} = 164 \text{ ft. forward of aft perpendicular.}$$

* *The vertical position is required for stability purposes, as afterwards mentioned.*

Suppose a design is being got out for a vessel to carry 4,640 tons deadweight on a draught corresponding to Lloyd's summer free-

board, and of 9 knots speed, the particulars shown in the foregoing table representing the vessel. Let the proposed dimensions be:

Length, B.P.	320 ft.
Breadth, Moulded	45 ft.
Depth, Moulded	23 ft.

The calculated freeboard being 3 ft. $2\frac{1}{2}$ in., the draught is obtained as shown in the recent example, where the figures for this vessel were used. The extreme load draught—*i.e.*, to the bottom of the keel—is 20 ft. $1\frac{1}{2}$ in. Above, the lightweight is given as 1,990 tons. This, added to the required deadweight of 4,640 tons, gives a load displacement of 6,630 tons, as is also shown. We have now to design the form of the vessel to give a displacement of 6,630 tons at 20 ft. $1\frac{1}{2}$ in. extreme draught. In designing the form the lines are drawn to the moulded dimensions—*i.e.*, to the inside of the plating. The plating itself contributes an amount of displacement, and since we are to design to the moulded form it is necessary to deduct from the total displacement, the displacement of the plating, as well as that of any other appendages, such as keel, bilge-keel, rudder, &c., so as to find the amount of *moulded* displacement required. For the shell and appendages in a vessel such as the above a deduction should be made of about ·7 per cent. of the total displacement:

$$6{,}630 \text{ tons} - \cdot 7 \text{ per cent.}$$
$$= 6{,}633 - 46{\cdot}4 = 6{,}583{\cdot}6 \text{ tons,}$$

which is the moulded displacement required. The draught to which this displacement is to be obtained is also to be *moulded*—*i.e.*, the extreme draught reduced by the thickness of the keel. In this case, where we have an extreme draught of 20 ft. $1\frac{1}{2}$ in. and a keel $1\frac{1}{2}$ in. deep, the moulded draught is 20 ft. We have now to design the moulded form of a vessel of the above dimensions to give 6,584 tons displacement at 20 ft. moulded draught. The block co-efficient for this would be:

$$\frac{6{,}584 \times 35}{320 \times 45 \times 20} = \cdot 8$$

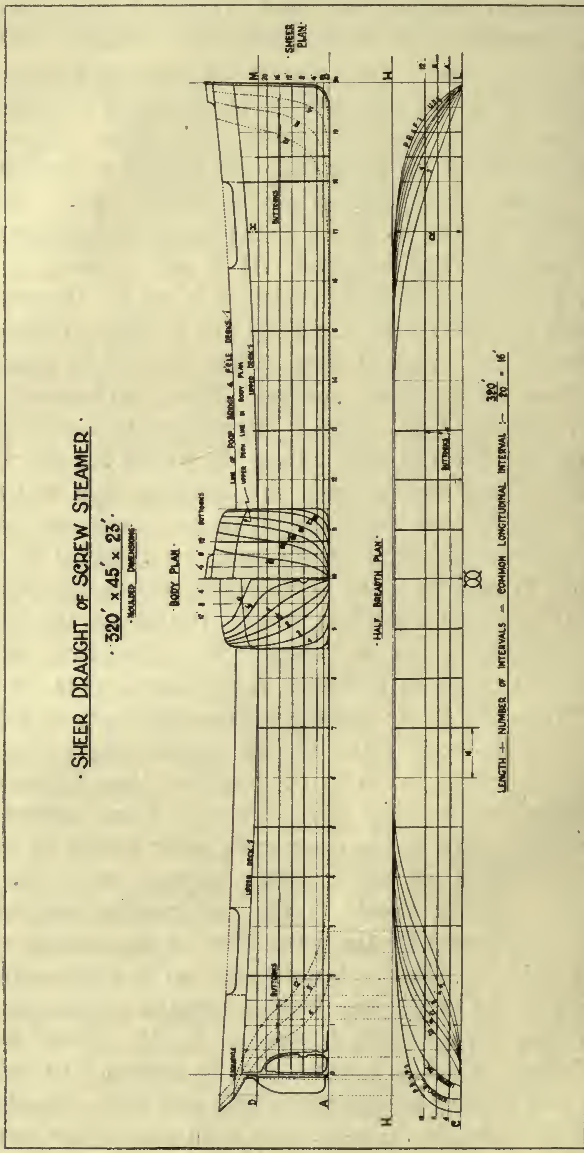
which, for the dimensions and required speed, is fairly suitable. Should the vessel be required to float at even keel when loaded, it will be obvious that the longitudinal position of the Centre of Buoyancy must be placed at 164 ft. forward of the aft perpendicular,

so as to be immediately under the position of the Centre of Gravity, which was calculated in the above. In other words, the centre of support is to be placed directly under the centre of the weight. The drawings of the vessel's form, giving the required displacement and Centre of Buoyancy, may now be proceeded with.

The Sheer Draught is the name given to the plans upon which the shape of the vessel's form is illustrated. The form is obtained and faired up by ordinary geometrical methods, using elevation plan and sections. The designing and fairing of a ship's lines in the Sheer Draught can be claimed to be the most beautiful and interesting problem in solid geometry. In ship work the elevation is termed Sheer Plan or Profile. In the present articles it will be called Sheer Plan. Fig. 19 shows the Sheer Draught for the above vessel. The dimensions are first of all laid off in block form as follows : A base line, A B, for the Sheer Plan is drawn, upon which the length of the vessel is measured and perpendiculars erected at each end. The length used is the length B P, one perpendicular, therefore, being the after side of the stern post, and the other the fore side of the stem, as explained in Chapter I. Measuring above this base-line, the depth moulded is set off, and a fine D M is drawn parallel to the base-line. This is termed the depth moulded line. We have now completed a rectangle which represents, in block form the length and depth of the vessel. At a convenient distance below another block is constructed, in which the plan view is drawn, this being known as the half-breadth plan. The vessel's centre line C L is drawn, and then the half-breadth moulded is set off from it (both sides of the vessel being alike, only one side need be drawn), and the half-breadth line H H is drawn. The plan in which the shape of the sections is shown is called the body plan, a block being next constructed for this. This plan usually has the same base-line as the sheer plan and is placed either at one end, clear of this plan, or at the middle of its length, as shown in Fig. 19. The centre line of the body plan being drawn, the breadth moulded of the vessel is set off and verticals erected. The depth moulded is also drawn in this plan, which line is already drawn if the body plan is placed at mid-length of the sheer plan. In the body plan is shown the shape of the vessel at various points, which are equally spaced throughout the vessel's length, these points being spaced off and perpendiculars erected at them in the sheer and half-breadth plans. It must here be decided where the displacement is to be measured from—*i.e.*,

the aft perpendicular or the fore side of the propeller aperture. In this case the length for displacement is taken from the aft perpendicular, and the sections spaced accordingly. The number of sections required is dependent upon whichever rule is to be employed in calculating the displacement, &c. Simpson's First Rule is used in most cases. The form of the stem and stern and the amount of sheer having been decided, the shapes of the displacement sections can now be sketched in the body plan. It is usual to sketch these sections by using the body plan of a previous similar ship for a guide, which enables one to obtain "lines" which will be nearly fair, as well as giving a displacement somewhat slightly more or less than that required. It is not, however, absolutely necessary to have another body plan as a guide; the sections can be sketched in by carefully using the eye to obtain fairness, as far as possible, in the shape of each section, as well as symmetry regarding their longitudinal spacing. These preliminary sections are now to be *faired up*. This is commenced by laying off water-lines in the half-breadth plan, lifting the widths for any particular water-line at each section from the corresponding water-lines in the body plan and setting them off on the respective sections in the half-breadth, and through the spots so obtained to draw the water line. It may be found necessary to depart from some of the spots so as to obtain a fair line, but by this means the form of the vessel is gradually *faired up*. After a few water-lines are laid off, one or two buttocks may be drawn in the sheer plan in the following method: Take the 12 ft. buttock in the after body of Fig. 19, for instance. In the body plan, lift the heights above the base-line at which this buttock is cut by the sections; these spots are shown by the black dots in that plan. Transfer these heights to their corresponding section in the sheer plan, again shown by black dots in Fig. 19. There are other points through which the buttock should pass. At the points in the half-breadth plan where this buttock is intersected by the water-lines we have spots, shown by the black dots, by which, when squared up to their corresponding water-lines in the sheer plan, further spots are obtained. This is shown by the dotted vertical lines in Fig. 19. These are all the points through which a buttock can be drawn. By working in this way and making modifications here and there, the three plans are gradually brought to agreement with each other, resulting in the *faired up* form of the vessel. The above brief description of the construction and fairing of the sheer draught, while not intended

Fig. 19.


SHEER DRAUGHT OF SCREW STEAMER
320' x 45' x 23'
MOULDED DIMENSIONS
BODY PLAN
SHEER PLAN
HALF BREADTH PLAN
LINE OF POOP BRIDGE & F'C'LE DECKS
UPPER DECK LINE IN BODY PLAN
UPPER DECK
BUTTOCKS
LENGTH ÷ NUMBER OF INTERVALS = COMMON LONGITUDINAL INTERVAL :- 320'/20 = 16'

as a treatise on laying off, shows the adopted means whereby the "lines" of a vessel are determined in that plan. This part of the subject can only be learnt and mastered by practical experience; therefore, to all students of this particular item of ship-design the actual construction of a sheer draught is recommended. The faired up form having been eventually determined, the next step is to calculate the displacement and the position of the centre of buoyancy longitudinally. This is done by finding the area of each section, and then by putting these areas through Simpson's rule to find the volume as was explained and done in connection with Fig. 6 in Chapter I. The area of each separate section can be first found by using Simpson's rule, but since this calculation is only of a preliminary nature, so as to find how the results obtained from the "lines" compare with the required displacement and centre of buoyancy, a quicker method is employed by means of the use of the planimeter, an instrument of great value in rapidly ascertaining areas.

Calculating the Displacement, etc. The planimeter being fixed in position, the *reading* of each half-section (bounded by the centre-line, frame-line and the water-line to which the displacement is required—20 ft. in this case) is obtained by tracing the pointer of the instrument around each required area. The planimeter readings having been obtained, they are then put through Simpson's rule for the purpose of finding the volume of displacement, also being multiplied by "levers" to find the "longitudinal centre of buoyancy." By use of the planimeter, the displacement and longitudinal centre of buoyancy, as represented by the "lines," are therefore quickly obtained. Of course, one cannot always expect to obtain the correct displacement and L C B at the first attempt, and from this preliminary calculation the required amount of alteration is ascertained, and the "lines" can be modified in accordance thereto when another planimeter calculation is made. In this way we eventually obtain the form that will fulfil the required conditions of displacement and longitudinal centre of buoyancy. After the "lines" are fixed according to the planimeter calculation, the final displacement sheet calculations may be commenced. For these calculations the widths of the water-lines are measured at the various sections, and then by use of Simpson's rules the areas of water-lines can be found, or by using the widths of the various water-lines at any particular section the area of that section can be found. If

the areas of water-lines or of sections as found in this way are then put through Simpson's rule, the volume of displacement can be found. It is usual to use both methods, working vertically with the water-line areas and longitudinally with sectional areas, and by use of "levers" and "moments" to find the position of the centre of buoyancy both vertically and longitudinally. The areas of the water-lines having been found, their centres of flotation (or centre of gravity of area) is then found by the use of "levers" and "moments." The tons per inch, being dependent upon the area of water-line, is also found at this stage by dividing the area by 420, as was seen in Chapter II. The area of 'midship section, being often required, is also calculated by using the water-line widths upon the displacement section at 'midships and putting them through Simpson's rule.

When dealing with questions of resistance, the area of the immersed surface is often required, and, therefore, this generally forms another branch of the present calculations. The displacement of the shell can also be found from the area of the wetted surface when multiplied by its mean thickness.

The following shows a sample calculation for wetted surface area and the shell displacement, the minor appendages being added. The method adopted is to take the half-girths of the sections and put them through the rule multipliers, and find the mean immersed half-girth by dividing the sum of functions so obtained by the

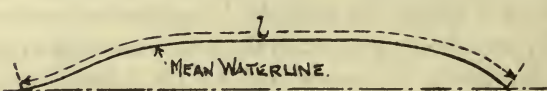


Fig. 20.

sum of the multipliers used. The half-girths are obtained from the body plan of the sheer draught or the model, by measuring round the outside of the section from the centre line at base up to the required water-line. The mean immersed half-girth so found being multiplied by the mean length of water-line and then by 2 for both sides, gives the total area of wetted surface. The mean length of water-line can be found in a similar way to that used in finding the mean half-girth of the sections, although it is quite near enough to take the length of a water-line at half the required draught. For instance, suppose that the given draught is 14 ft. Let the water-line shown in Fig. 20 be one at half of this draught—

viz., 7 ft. Measure round the outside of this water-line, as shown by 1, and use this length to obtain the wetted surface.

WETTED SURFACE UP TO 14 FT. W L.

No. of Section	Half-girths	Simpson's Multipliers	Functions
0	14·00	1	14·00
1	22·50	4	90·00
2	30·80	2	61·60
3	22·60	4	90·40
4	14·00	1	14·00

Sum of multipliers = 12) 270·00

22·5 ft. = mean immersed half-girth.

Mean length of water-line
= 304 ft.

6,840
2 sides.

13,680 sq. ft. wetted surface.

Mean thickness of
Shell × $1\frac{1}{2}$ = ($\frac{10}{20}$in. × $1\frac{1}{2}$ = $\frac{3}{4}$in.) = ·06 of a foot.
($\frac{10}{20}$ = thickness of plates—

820·8 cb. ft. displacement of shell.

+ Rudder, propeller and
bilge keels = 19·2 ,, ,,

35) 840·0

24 tons displacement of shell and minor appendages.

NOTE.—This calculation does not refer to the vessel represented in Fig. 19.

It is only when the wetted surface is being calculated for the various draughts that the shell displacement is found in the manner shown above. When no wetted surface calculation is being made the amount of shell displacement is taken as being a percentage of

the moulded displacement, the following being good figures to use for such :

	Load Draught	Light Draught
Fine vessels (about ·5 block co-efficient) ...	1·00%	2·0%
Full vessels (about ·8 block co-efficient) ...	·65%	1·5%

However, when the areas of wetted surface are being calculated, advantage may be taken to calculate from this the shell displacement, as shown in the foregoing example.

The calculations for the different items, as mentioned above, are made for a number of different draughts, and the results so obtained are set off in diagrammatic form in the displacement scale, as described in the following paragraph.

The "Displacement Scale" and the various Curves shown thereon. In Fig. 21 we have shown the displacement scale and the other curves as constructed from the results obtained by means of the before-mentioned calculations. Vertically, we have the draught scales from which the results are set off in a horizontal direction at their respective draughts. Two scales are shown—one being the moulded draughts measured above the base-line, which is the top of keel; the other being the extreme draughts, taken from the bottom of the keel. The calculations being made to the moulded draughts, as used in the sheer draught, the results must therefore be set off at these draughts in the displacement scale. The water-lines used in the calculations, 2 ft., 8 ft., 14 ft. and 20 ft., up to which the various results were obtained, are drawn across the diagram at right angles to the draught scale, as shown dotted in Fig. 21. The curve of displacement is first laid off by taking the displacement, as found in the calculation, at each of the water-lines and measuring it to a suitable scale, on the horizontal lines drawn at the respective draughts. Through these spots a curve is drawn, the scale used being shown at the top of the diagram. This curve is extremely useful in obtaining the displacement at any particular draught, or the draught corresponding to any given displacement. For instance, suppose this vessel to be floating at a draught of 15 ft. 6½ in. forward and 16 ft. 9½ in. aft, and it is required to find the displacement corresponding to this condition, the mean draught is first found :

F	15 ft.	6½ in.
A	16 ft.	9½ in.
	2)32 ft.	4 in.
Mean draught =	16 ft.	2 in.

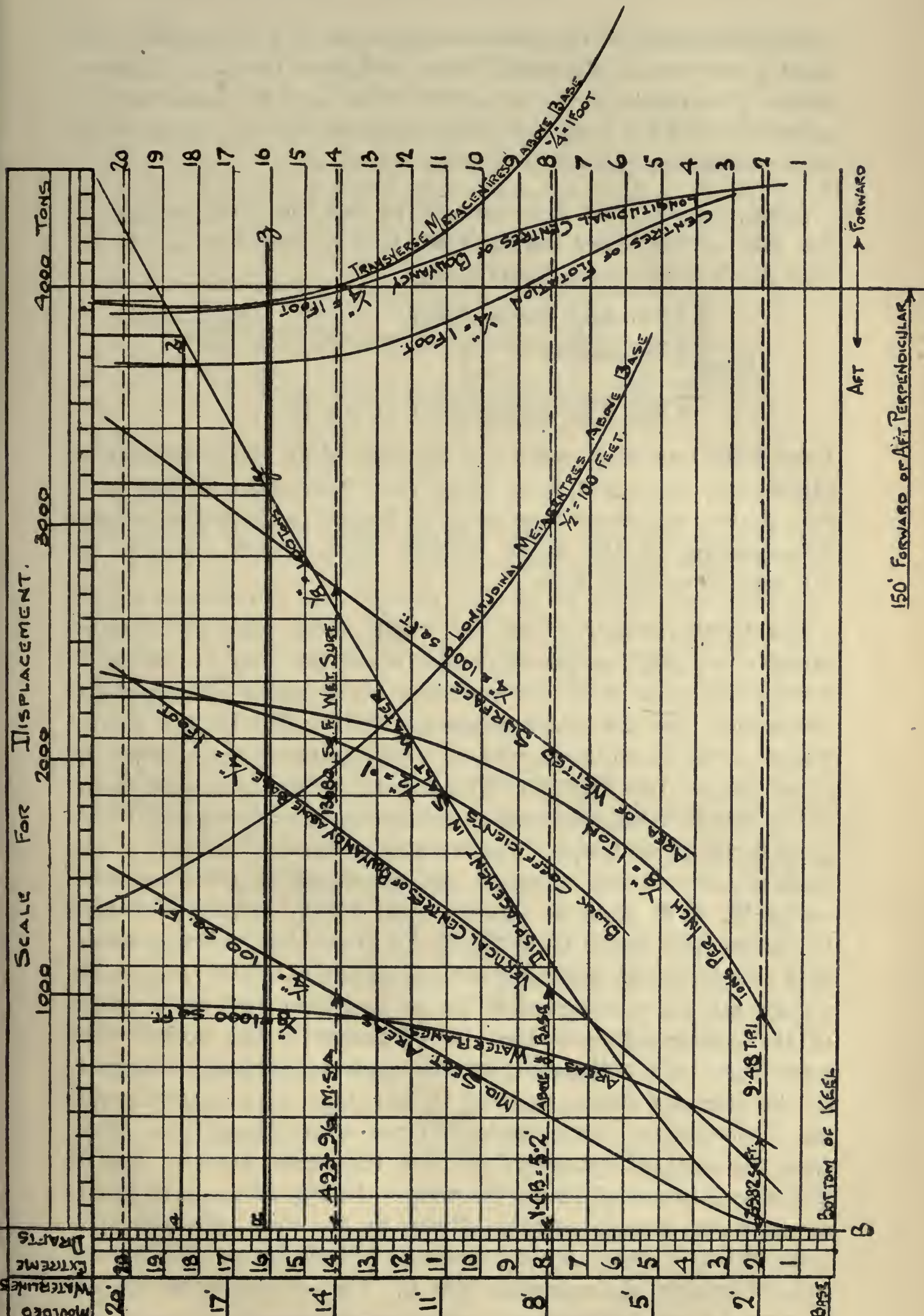

SCALE FOR DISPLACEMENT. TONS
1000
2000
3000
4000
TRANSVERSE METACENTRES ABOVE BASE
LONGITUDINAL CENTRES OF BOUYANCY
CENTRES OF FLOTATION
LONGITUDINAL METACENTRES ABOVE BASE
½" = 100 FEET.
AREA OF WETTED SURFACE
TONS PER INCH
BLOCK COEFFICIENTS
DISPLACEMENT IN SALT WATER
VERTICAL CENTRES OF BOUYANCY ABOVE BASE
AREAS WATER PLANES
MID. SECT. AREAS
13680 SQ. F. WET SURF.
423·96 M.S.A.
V.C.B. = 5·2' ABOVE BASE
9·48 T.P.I.
BOTTOM OF KEEL
150' FORWARD OF AFT PERPENDICULAR
AFT
FORWARD
MOULDED WATERLINES
EXTREME DRAFTS
BASE

Fig. 21.

At this draught on the extreme scale, a line **x y** is squared across until it cuts the displacement curve, and from the point of intersection **y**, a perpendicular is erected which cuts the scale for displacement at 3,170 tons, this being the displacement corresponding to the mean draught of 16 ft. 2 in.

Again, suppose that it is required to find the mean draught at the time of the vessel having 2,300 tons of cargo on board, her light displacement being 1,500 tons.

1,500	light displacement.
2,300	cargo.
3,800	total displacement at the time.

From 3,800 tons displacement in the scale, draw the perpendicular cutting the curve at **b**, from which point the horizontal line **ba** is next drawn, and which intersects the draught scale at the draught corresponding to the above amount displacement. In Fig. 21 it is seen to be 18 ft. 6 in.

The figures obtained from the displacement scale for a mean draught are fairly accurate except when the trim is excessive. Taking the mean of the forward and aft draughts, we make the assumption that the actual inclined water-line $\mathbf{W}^1$ $\mathbf{L}^1$ and a level water-line **W L** each giving the same displacement) are intersecting at 'midships. (*See* Fig. 22.) This may not be the case, as is shown in this sketch, where these two water-lines are intersecting at **F**, which is 10 ft. aft of 'midships. For the displacement to be equal in the inclined and level conditions, the amount contained in the immersed wedge $\mathbf{W}^1$ **F W** must be exactly equal to the amount contained in the emerged wedge $\mathbf{L}^1$ **F L**. If this is the case, and the water-lines do intersect at 'midships, the assumption is correct; but should we assume the intersection to be at 'midships, and the contents of the corresponding wedges—as produced by the dotted level water-line—be not equal, then the result is obviously incorrect. In the case of a vessel trimming by the stern at the load draught, the displacement, corresponding to the mean draught, as taken from the scale, is generally less than the actual amount, due to the reasons explained in the following: In Fig. 22 we have shown a vessel with a large amount of trim by the stern, the water-line being $\mathbf{W}^1$ $\mathbf{L}^1$. With an equal amount of displacement, but floating at level draught, the water-line is **W L**. The volumes of the wedges

W^1 **F W** and **L**^{1} **F L** are equal since the displacement has not changed. The end draughts being 12 ft. and 7 ft., we have a mean at 'midships of 9 ft. 6 in. Setting this off at 'midships, as shown in the sketch, it will be seen that we obtain a different water-line (shown dotted) to the actual level water-line **W L**, which is equal to 10 ft. draught, and cuts the inclined water-line **W**^{1} **L**^{1} at **F**.

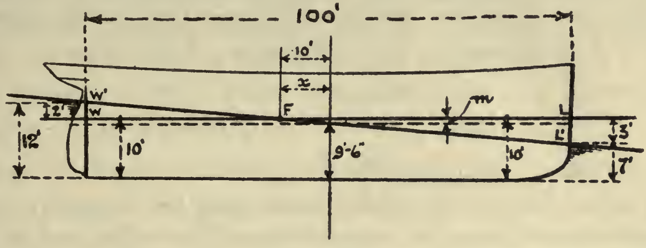


Fig. 22.

The "mean draught displacement" would therefore be less than the actual by the amount contained between the water-lines at 9 ft. 6 in. and 10 ft. level draughts. To look at this in another way, suppose that the vessel originally floats at 10 ft. mean draught, and then, on account of a weight being shifted the vessel changes trim, going down by the stern and up by the head. The point at which the water-lines intersect is approximately **F**, the "centre of flotation"—sometimes called the "tipping centre"—which, on account of being in the after body in this case, has the effect to increase the draught aft by 2 ft. and reduce the draught forward by 3 ft., the difference in these amounts being due to the position of the centre of flotation. The draughts now become 12 ft. aft and 7 ft. forward, which gives a mean of 9 ft. 6 in. Taking a displacement from the scale for this draught of 9 ft. 6 in. would obviously be incorrect, since the inclined displacement is one of the same amount as when floating at 10 ft. mean draught. In this case the difference is large, being about 7 per cent., and should be taken account of in the following manner. Take the mean of the new draughts :

	12 ft.	0 in.
	7 ft.	0 in.
2)	19 ft.	0 in.
=	9 ft.	6 in.

and to this add the distance m between the two level water-lines as found by:

$$m = x \times \frac{\text{Total trim}}{\text{Length of water-line}}$$

where x is the distance of the centre of flotation aft of 'midships, the total trim being the difference between the new draughts, 12 ft. aft — 7 ft. forward = 5 ft. total trim. In this case—

$$m = 10 \text{ ft} \times \frac{5 \text{ ft.}}{100 \text{ ft.}} = \cdot 5 \text{ ft.} = 6 \text{ in.}$$

9 ft. 6 in. + 6 in. = 10 ft., the latter being the draught at which the displacement may be measured from the scale, and in most cases it will be very nearly correct. If the centre of flotation were in the fore body and the vessel trimming by the stern, the displacement corresponding to a mean of the draughts will be in excess of the actual; therefore the correction for m is deducted in such a case so as to obtain the level draught at which the displacement corresponds to that of the inclined condition:

Trim by	stern and C F	aft	of 'midships.	—add **m**.
,,	,,	,, forward	,,	.—deduct **m**.
,,	head	,, aft	,,	.—deduct **m**.
,,	,,	,, forward	,,	.—add **m**.

The difference of displacement when using a mean draught is only worth taking into account when the trim is large and the centre of flotation far from 'midships; for instance, in the case of an ordinary cargo vessel 300 ft. long, with the centre of flotation 5 ft. aft of 'midships, and having 5 ft. of trim by the stern, the difference in displacement would be about 25 tons, an amount certainly worthy of notice.

In the case of an extremely large amount of trim, as for example, that of a small coasting steamer with engines aft, and in the light condition, where the draughts may be about 11 ft. aft and 1 ft. forward, the displacement scale cannot be expected to give the displacement satisfactorily if the mean draught is taken, and therefore a separate calculation should be made with the planimeter, although by means of the above method a fairly near result may be obtained. From the above we see that in the case of vessels

that are to have much trim, it is very necessary to design their lines by working parallel to the required trim, and to make the displacement and other calculations from the same.

The curve of block co-efficients is generally drawn as shown in Fig. 21.

Curve of Longitudinal Centres of Buoyancy. A perpendicular is erected to represent 'midships, and the positions as calculated are set off from this on the respective water-lines, and a curve drawn as shown, from which the position for any intermediate water-line can be measured.

Curves of Vertical Centres of Buoyancy, etc. The vertical positions of the C B being found relative to the base line, these distances are now set off in the diagram by measuring them along the dotted water-lines for the respective draughts ; for instance, at the 8 ft. water-line the C B is 5·2 ft. above base ; upon this water-line the amount is measured off from the perpendicular A B of the draught scale, as shown by the arrow marks. For the curves of areas of waterplanes, tons per inch, areas of 'midship section, and areas of wetted surface, the results are set off from the perpendicular A B. Offsets for area of waterplane and " tons per inch " are shown upon the 2 ft. water-line ; and for the 'midship section area and wetted surface, offsets are shown on the 14 ft. water-line. The curve of centres of flotation is laid off in the same way as described for the longitudinal centres of buoyancy. It should be noted that in laying off the curves the results are set off at moulded draughts, to which the calculations are made. These draughts are shown by the dotted lines in Fig. 21.

The curves shown upon the displacement scale are extremely useful in enabling one to obtain particulars of the vessel at any required draught.

CHAPTER IV.

Initial Statical Stability: Conditions. Transverse Metacentre Explained. Formula for Calculating the Position of the Transverse Metacentre. Average Values of Metacentric Heights. The Metacentric Diagram; its Construction and Use.

Conditions of Stable Equilibrium. We have already seen in Chapter II. that *for a vessel to be floating in equilibrium in still water she must displace an amount of water the weight of which is equal to the weight of the vessel, and also that the centre of gravity must lie in the same vertical line as the centre of buoyancy.* This is a condition of equilibrium, and by adding: "*And the centre of gravity must be in such a position (below the metacentre) so that if the vessel were were inclined, the forces of gravity and buoyancy would tend to bring the vessel back to her former position of rest,*" we then have the condition of stable equilibrium. *Stable equilibrium* means that when a vessel is inclined from the upright she will return to that position again as soon as the inclining force is relaxed. *Unstable equilibrium* means that when a vessel is inclined from the upright she will not return to that position, but incline further away from it. *Neutral equilibrium* means that when a vessel is inclined from the upright she will neither return to the upright nor incline further away from it, remaining in the position she is inclined to. The first portion of the above conditions is simply a condition of equilibrium, and holds good in the case of neutral as well as stable equilibrium, since in both we have the C G and C B in the same vertical line. It can even be said to apply to a vessel in a condition of unstable equilibrium while she is in the upright position, since we would again have the C G and C B lying on the same vertical line, but this would be only *momentary equilibrium*, and an impracticable position. In all these cases the vessel's centre of gravity is assumed to lie on the centre line.

In Figs. 23, 24 and 25, which are transverse sections of a vessel, we have the three conditions represented, an upright and inclined position being shown for each condition. Upon being inclined, the centre of buoyancy B shifts out to B_1, the centre of gravity G remaining fixed. In Fig. 23, where the condition of stable equilibrium

is shown, it will be seen that when the vessel is inclined, the downward force of the weight acting through G and the upward force of the buoyancy acting through B_1 are acting with a "couple," G Z,

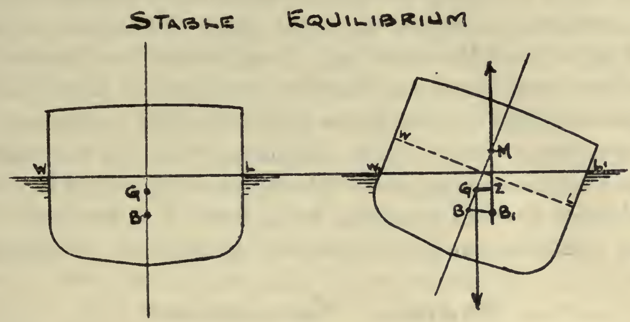


Fig. 23.

which tends to take the vessel back again to the upright. The line of the upward force of buoyancy cuts the centre line of the vessel at a point M, which is known as the Transverse Metacentre. It should be here noted that M is above G, thereby fulfilling the latter portion of the condition of stable equilibrium as already given. Fig. 24 represents the condition of unstable equilibrium and it will be noticed that the forces are acting with a "couple," Z G, in directions which tend to incline the vessel further from the upright. In this case M is below G. The upright position shown in this figure is the condition of momentary equilibrium, since we here have G and B in the same vertical line; but on the slightest inclination we obtain an upsetting moment, and since it is impossible to

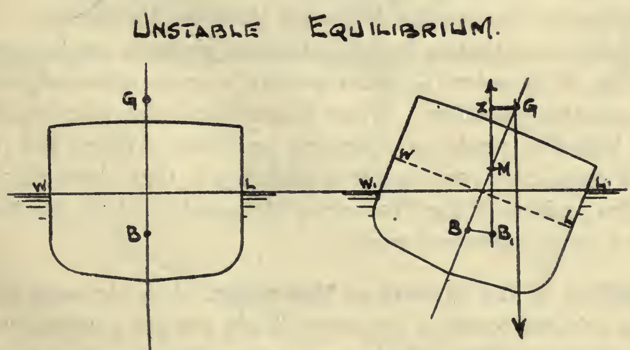


Fig. 24.

keep the vessel rigidly upright, the condition is therefore an unstable one. In Fig. 25 we have an example of neutral or indifferent equilibrium, since here the vessel possesses neither a "righting couple" nor a "couple" that will capsize her, because the downward and upward forces are acting in the same vertical line, this being caused by G and M coinciding. It will have been noticed that, in the first two cases, the vessel's stability was dependent upon the distances between the forces of gravity and buoyancy, which is known as the "lever" and designated G Z. In the stable condition we have a righting lever, M being above G; and in the unstable condition we have an upsetting lever, since M is below G. In the neutral condition we have neither a righting nor upsetting lever,

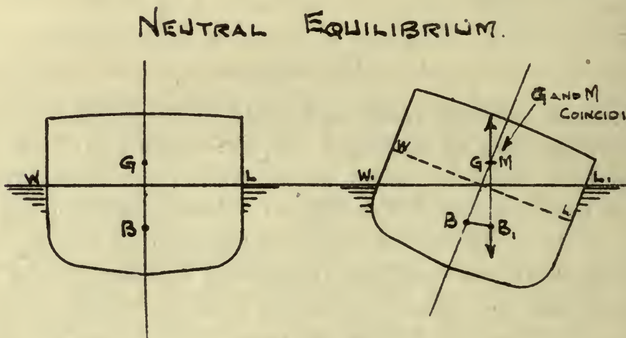


Fig. 25.

since the upward force of buoyancy intersects the vessel's centre line at G, so that we have the position of M coinciding with G. It will be seen that the positions of G, the centre of gravity, and M, the metacentre, govern the lever and therefore the stability. When M is above G, a righting lever is obtained giving a stable condition; and when M is below G, an upsetting lever is obtained, resulting in an unstable condition. When M and G coincide there is no lever, which therefore produces a neutral condition. When the position of G is fixed, the vessel's initial stability is then depending solely upon the position of the Transverse Metacentre, which, we therefore see, is a most important point.

Definition of the Transverse Metacentre. *It is the point vertically in line with the centre of buoyancy in the upright position, which is also vertically in line with the new centre of buoyancy, corresponding to a small transverse inclination to the upright. It is also the point*

above which the centre of gravity must not be raised for the vessel to remain in stable equilibrium. The distance between G and M is the "metacentric height," generally spoken of as G M, being positive when M is above G, and negative when M is below G. The position of the Transverse Metacentre for the upright condition is found by dividing the "transverse moment of inertia of waterplane" by the volume of displacement up to that waterplane, the result being the distance of M above the corresponding centre of buoyancy, the amount being spoken of as B M. We therefore see that a large moment of inertia and a small displacement give a high position of M, a large moment of inertia being obtained by having a large width of waterplane. During inclination, the position of M changes, because the moment of inertia has been changing, but for all practical purposes it may be said that in most ships its position remains constant up to inclinations of 10 and 15 degs., and sometimes even further; therefore, within these limits, we can give consideration to the vessel's stability by basing it solely upon the amount of "metacentric height."

Formula for calculating the position of the Transverse Metacentre. Let Fig. 26 represent a vessel transversely inclined over to a small angle θ (exaggerated for clearness in the sketch); W F W_1 is the wedge of emersion and L^1 F L is the wedge of immersion. It will

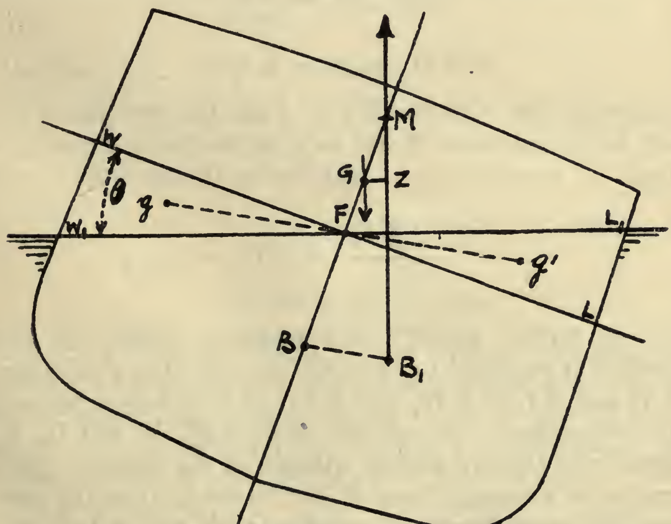


Fig. 26.

be obvious that their areas must be equal so as to allow of the vessel retaining the same amount of displacement. Areas of the section are used for the sake of simplicity; the vessel is supposed to be prismatic, so that we need take no account of the length; the areas therefore, represent volumes. Let—

v = the area of either wedge. (Representing volume.)
g = the centre of gravity of emerged wedge.
g_1 = the centre of gravity of immersed wedge.
B = the centre of buoyancy in upright position.
B_1 = the centre of buoyancy in inclined position.
V = the total immersed area of the vessel's section. (Representing Volume.)
M = the Transverse Metacentre.

On looking into Fig. 26 it will be seen that on account of the inclination and the resultant underwater form of the vessel, B has shifted to B_1, and a vertical drawn through this point, at right angles to the new water-line $W_1 L_1$, cuts the former vertical—*i.e.*, the vessel's centre line—at M, the Transverse Metacentre. If we obtain the value of $B B_1$, we can find B M, because at small angles of inclination

$$B M = \frac{B B_1}{\tan \theta}.$$

$$\text{or } B M \tan \theta = B B_1$$

We therefore first of all find $B B_1$. From the principle of moments it will be obvious that B will shift in the same direction as the wedges—*i.e.*, parallel to $g g_1$—and that its amount will be equal to—

$$\frac{v \times g g_1}{V} = B B_1$$

Let us further simplify this equation. Since the angle of inclination is supposed to be small, we may take y as being equal to F W and F L or F W_1 and F L_1, which are each half-ordinates of the waterplane. We can also assume W_1 W and L_1 L to be straight lines, thereby making triangles of the wedges. The centre of gravity of a triangle being at $\frac{2}{3}$ of its height from the apex, g F. which also equals F g_1, will therefore equal $\frac{2}{3}y$, or g g_1 will equal $\frac{4}{3}y$.

The volume of a wedge—

$$v = \frac{y \times W_1\, W}{2}\ \text{(emerged wedge)} \quad \text{or} \quad \frac{y \times L_1\, L}{2}\ \text{(immerged wedge)}$$

being simply the area of the triangles.

Since $W\,W_1$ and $L\,L_1 = y \tan \theta$, we may write—

$$v = \frac{y \times y \tan \theta}{2} = \tfrac{1}{2}\, y^2 \tan \theta.$$

Having now found the values of v and $g\,g_1$, we can write—

$$B\,B_1 = \frac{v \times g\,g_1}{V} = \frac{\frac{1}{2}\, y^2 \tan \theta \times \frac{4}{3}\, y}{V}$$

$$= \frac{\frac{2}{3}\, y^3 \tan \theta}{V} = B\,B_1$$

The angle being small, so that $B\,M \tan \theta = B\,B^1$,

$$\therefore B\,M \cancel{\tan \theta} = \frac{\frac{2}{3} y^3 \cancel{\tan \theta}}{V}$$

$\tan \theta$ cancelling out, we have remaining—

$$B\,M = \frac{\frac{2}{3}\, y^3}{V}$$

which is equal to—

$$B\,M = \frac{\text{Moment of Inertia}}{\text{Volume of Displacement}}$$

$\frac{2}{3}y^3$ is equal to $\frac{1}{3}$ of cubes of half-ordinates multiplied by 2 for both sides, which gives the transverse moment of inertia of the waterplane about a longitudinal axis passing through its centre line. Knowing the value of B M, we next find the height of B above the base, and adding this to B M, we have the height of M above base.

If, from the height of M above base, we take the height of G above base, we obtain G M, the Metacentric Height.

Values of Metacentric Heights. The following are average figures giving good results in working conditions.

	Ft.	Ins.		Ft.	Ins.
Cargo steamers	0	9	to	1	6
Sailing vessels	2	6	to	3	6
High speed liners	1	0	to	2	0
Fast channel steamers	1	3	to	2	0
Paddle excursion steamers	1	6	to	2	6
Tugs	1	0	to	2	0
Steam yachts	0	9	to	1	9
Battleships	3	0	to	3	6
Cruisers	2	0	to	2	6
Destroyers	1	9	to	2	6

In the designing stages, or at other times when it is impossible to obtain the calculated value of B M, it can be approximated by the formula obtained as follows :

$$B\,M = \frac{\text{Moment of Inertia}}{\text{Volume of Displacement}}$$

$$= \frac{\cancel{L} \times \cancel{B^3}^{\,2} \times i}{\cancel{L} \times \cancel{B} \times D \times c_b} = \frac{\text{Approximate Moment of Inertia}}{\text{Volume of Displacement.}}$$

$$\text{Cancelling L and B} = \frac{B^2 \times i}{D \times c_b} = \frac{B^2}{D}\,x.$$

i is a co-efficient which when multiplied by $(L \times B^3)$ gives the M I.

c_b is the block co-efficient of fineness.

x is the co-efficient obtained by combining i and c_b.

Therefore we may write—

$$\frac{\text{Breadth}^2}{\text{Draught}} \times x = \text{Approximate B M.}$$

An average value for x is ·08.

The Metacentric Diagram is constructed from the results of calculations made for the value of B M at a few draughts. The following shows a calculation made for the transverse B M. In the proof of the formula for B M we saw that $\frac{1}{3}$ of the cubes of half-ordinates of the waterplane, when multiplied by 2 for both sides and divided by the volume of displacement, would give the B M at the given waterplane. This method will be noticed in the following where the half-ordinates are first cubed and then put through Simpson's multipliers, the sum of functions of cubes being multiplied by $\frac{1}{3}$ of the longitudinal interval so as to complete the rule, and then by $\frac{1}{3}$ and 2 as required by the B M formula, the result being the transverse moment of inertia of the waterplane about a longitudinal axis passing through the centre of the vessel. This is next divided by the volume of displacement, as taken from the displacement calculation, which then gives the value of B M—*i.e.*, the height of the metacentre above the centre of buoyancy.

TRANSVERSE METACENTRE.

No. of Section.	8 ft. Water-line.			
	Half-Ords.	Cubes.	S. M.	Functs. of Cubes.
0	0·00	0	1	0
1	14·60	3,112	4	12,448
2	21·00	9,261	2	18,522
3	14·58	3,099	4	12,396
4	0·00	0	1	0
				43,366
	$\times$ $\frac{1}{3}$ of Longitudinal Interval			25
	$\times$ $\frac{1}{3}$ of Cubes			3)1,084,150
				361,383
	For Both Sides $\times$			2
	Transverse Moment of Inertia			722,766

$$\text{Volume of Displacement} = \overset{\text{Tons.}}{1{,}136} \times 35 = 39{,}760 \text{ cu. ft.}$$

$$\frac{I}{V} = \frac{722{,}766}{39{,}760} = 18{\cdot}18 \text{ ft. B M}$$

$$+ \text{ B above Base } 5{\cdot}20 \text{ ft.}$$

$$\text{M above Base} = 23{\cdot}38 \text{ ft.}$$

The moulded widths of the waterplane and moulded displacement are used for this calculation. Having found the position of the metacentre at a few draughts, as described and shown above, they are next laid off in diagrammatic form so that the position of M for any particular draught may be readily ascertained. In Fig. 27, a metacentric diagram is shown. The perpendicular A B is first

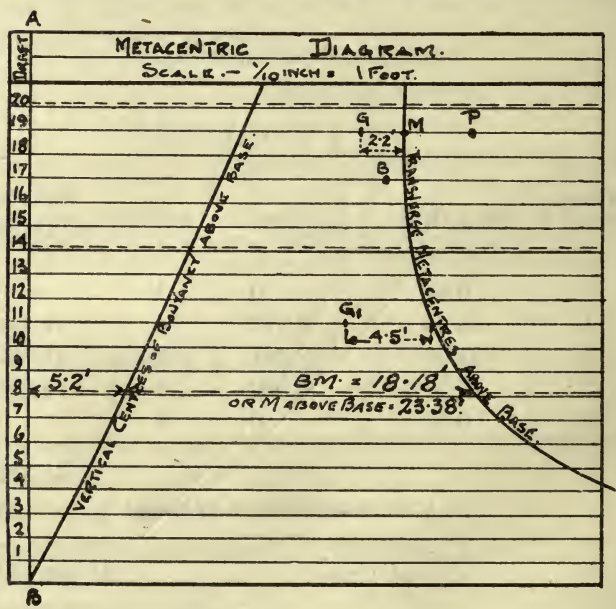


Fig. 27.

drawn, and upon it the scale of draughts is set off. At right angles to this perpendicular the values, as calculated, are set off upon lines drawn at the draughts corresponding to the calculations; these are shown dotted at the 8 ft., 14 ft., and 20 ft. water-lines.

As in the case of the displacement calculation and scale, these are moulded draughts, and, therefore, are situated higher than the extreme draughts. It is usual to put the curve or vertical centres of buoyancy upon the diagram. The curve of metacentres is next laid off by measuring the B M's (*i.e.*, the metacentre above the centre of buoyancy) from the curve of V C B's, as shown by the amount 18·18 ft. on the 8 ft. water-line, or 23·38 ft., which is the distance of M above base, can be measured from the perpendicular A B. The calculated positions of M having been set off, the curve can then be drawn. The diagram is completed by drawing across it level lines at the various draughts, and we then have a means of quickly obtaining the height of the Transverse Metacentre at any draught. The value of M above base can be obtained by measuring the full distance from the draught scale to the curve, or if the B M is required, the distance between the two curves is measured. The metacentric diagram, which gives a graphic representation of the locus of V C B's and Transverse Metacentres, is extremely useful to the naval architect or ship's officer, as from it they are able readily to obtain the position of the Transverse Metacentre at any particular draught, which position, when used in conjunction with the position of the Centre of Gravity, then enables them to state the conditions of the vessel's initial statical stability—*i.e.*, by obtaining the Metacentric Height G M. As an example of its use, take the case of the vessel dealt with in Chapter III., where a table was drawn up and the vertical position of the centre of gravity calculated. When loaded, the vertical height of C G was found to be 17·42 ft. above base, and when light 16·62 ft. above base. Assume that the curve of transverse metacentres shown in Fig. 27 is for this particular vessel, and say that when loaded the draught is 19 ft., and when light 11 ft., these assumptions being made for the sake of this example. Take the load condition first: At 19 ft. draught, set off the height of the centre of gravity 17·42 ft., giving the point G as shown in the diagram. Measuring the distance between this point G and the curve of metacentres, we find the metacentric height G M to be 2·2 ft.: for the light condition, the height of the centre of gravity, 16·62 ft., is set off at 11 ft. draught, giving the position G_1, and the resultant metacentric height of 4·5 ft. In both cases G is below M, and the vessel therefore is in a stable condition; but should it have occurred that, say for the load draught, the centre of gravity had been in the position as shown by P, so that it was above the metacentre, then the vessel would have been found to

be in an unstable condition when loaded, and therefore the disposition of the cargo would have to be so amended as to bring the centre of gravity below the curve. Should this occur in the designing stages (when the curve of transverse metacentres would be approximated from a previous similar ship by proportioning the B M's according to the square of the breadths of the two vessels—

$$\text{B M of new vessel} = \text{B M of previous similar vessel} \times \frac{B_n^2}{B^2}$$

where B_n = the breadth of the new vessel, and
B = the breadth of the previous vessel),

and should it be impossible to amend the disposition of the specified cargo, the vessel would then require to be altered in dimensions so as to obtain stability by means of increasing the breadth and reducing the depth. The added breadth would cause an increase of the moment of inertia of waterplane, and, consequently, the height of the metacentre would be increased. Reducing the depth would reduce the height of C G, and the combined effect of the two alterations would tend to give the vessel stability by changing the metacentric height from a negative to positive quantity. In designing a vessel where the stability is an important point, the metacentric height must be carefully looked into before the " lines " are fixed. The approximate curve, mentioned above, is admissible at the stage of fixed dimensions, but after the " lines " are drawn out, the metacentric diagram should be quickly constructed so as definitely to decide the question before building is actually commenced. It may not be necessary to draw out the diagram if the worst possible condition of the vessel's loading is known, as the metacentre may be calculated for this condition alone by using the ordinates of the corresponding waterplane and finding the B M in the usual way. The positions of the corresponding vertical centres of buoyancy, however, may not be known at this stage unless the displacement, etc., calculations are already made; if not, this position must be found by an approximate method, such as comparing it with another vessel whose " lines " are similar, taking the position above the keel as a proportion of the draught, the proportion used being obtained from the similar ship. In vessels of full form the centre of buoyancy above the base is equal to about ·55 of the moulded draught, and for finely shaped vessels ·6 can be used when the position is wanted roughly, though, of course, for

our present purpose the above-mentioned proportion must be used. Whichever method is used, the figure obtained for the V C B above base would not, however, be far from the correct one, and this then being added to the B M, gives the height of M above base. In the case of actual vessel, where the effect of a proposed cargo is being investigated, the only remaining course when a negative metacentric height is found to be the result, is to alter the position of the centre of gravity by placing the heaviest weights in the lowest positions, thus reducing the height of G. If the cargo is a homogeneous one, the only means left is to fill a bottom ballast tank, the addition of which will mean the reduction of an equal amount of cargo if the draught is not to be increased, the combined effect being to provide the vessel with stability. Another way in which the stability of a vessel may be altered is by the burning out of the bunkers. Take the case just mentioned, where a vessel at 19 ft. draught has 2·2 ft. G M, and suppose an amount of bunkers is burnt out, so that the draught is reduced to 17 ft. If the bunkers are situated in a low position, the effect will be to allow the centre of gravity to rise, since the reduction of weight is taking place in the low position. Suppose that in this case where G in the load condition is 17·42 ft. above base, the effect of burning out the bunkers to the above-mentioned extent is to increase the height of G by 1 ft. 3 in., which now gives G at 18·67 ft. above base, as shown by B in Fig. 27. The metacentric height is now only 1 ft. against 2·2 ft. in the fully load condition. This being a large reduction, shows that but for the vessel having a fairly large amount of G M in the original condition, she might have been left with a dangerously small amount, or even in an unstable condition. The metacentric diagram is therefore seen to be a valuable asset to the ship's officer, who is nowadays, in some cases, supplied with it. The metacentre being solely dependent upon the vessel's form, it is therefore fixed for any draught, the stability being then dependent upon the disposition of the weights. Having the diagram, it then remains to estimate the position of G for the particular condition of loading, and knowing the position of G for the light condition, the cargo, bunkers, etc., are then added, and by the principle of moments the combined centre of gravity for the total weight is found, the positions of the cargo, etc., being obtained from the capacity plan supplied to the vessel. The condition of the vessel can therefore be obtained before loading is commenced, or by these means the positions of cargo can be obtained so as to produce any required condition, or again,

it can be ascertained if ballasting will be necessary for any particular disposition of cargo. A large metacentric height produces a "stiff" vessel and one that will roll heavily; this could be avoided by making the above investigations and fixing an amount so as to obtain a safe quantity of stability without making the vessel excessively stiff. The following is an example of such an estimate for stability as made previous to the loading of a vessel, the "levers" being the distances of centres of gravity of the various quantities above the base:

Items.	Weight in Tons.	Levers in Feet.	Moments Foot-tons.
Vessel in Light Condition	1,000	15	15,000
Bunkers in Hold	100	10	1,000
Bunkers in Bridge	100	22	2,200
Cargo in Fore Hold	860	10	8,600
Cargo in After Hold	800	11	8,800
Fresh Water in Tanks in Bridge ...	10	22	220
Stores in Poop	5	24	120
	2,875		35,940

35,940 ÷ 2,875 = 12·5 ft. = Height of G above base.
Say 12·0 ft. = Height of M above base.

G would therefore be ·5 ft. above M, and the vessel in an unstable condition. Should the above quantities and positions of cargo be imperatively fixed, the only remaining means of providing stability is by ballasting. Say 400 tons of water is added in the double-ballast bottom tanks, the effect is found as follows:

	Weight in Tons.	Levers in Feet.	Moments Foot-tons.
Vessel in the above condition ...	2,875	12·5	35,940
Ballast in double-bottom tanks ...	400	1·5	600
	3,275		36,540

36,540 ÷ 3,275 = 11·14 ft. = Height of G above base. The addition of 400 tons will sink the vessel down to a deeper draught, at which the position of M will have changed; therefore this new position, for the new draught, must be taken from the metacentric diagram.

Say at the new draught the height of M is 11·75 ft. above the base.
G as altered by the ballast = 11·14 ft. ,, ,,
which gives a positive G M of ·61 ft.,

the vessel being therefore in a stable condition.

CHAPTER V.

METACENTRIC STABILITY. EFFECT OF INCLINATION AND SUCCESSIVE METACENTRES. TO DETERMINE PRACTICALLY, BY INCLINING EXPERIMENT, THE HEIGHT OF A VESSEL'S CENTRE OF GRAVITY; THE PROCEDURE AND NECESSARY PRECAUTIONS. EXAMPLE OF INCLINING EXPERIMENT.

Metacentric Stability. In Fig. 26 it will be seen that the righting lever G Z is equal to G M sin θ. It is usual to speak of statical stability as a "moment" of foot-tons, it being equal to the weight of the vessel, in tons, multiplied by the "righting lever" measured in feet. The weight being equal to the displacement and the righting lever equal to G Z, we therefore have the

$$\text{Moment of statical stability} = W \times G\,Z,$$
$$\text{or } W \times G\,M \sin \theta,$$

W being the displacement in tons and θ the angle of inclination. As previously mentioned, the position of M can be assumed to remain constant, in most cases, up to angles of 10 to 15 degrees; therefore, within these limits it will be seen that the amount of statical stability can be determined from the metacentric height, and is consequently spoken of as metacentric stability.

Effect of Inclination. As a vessel inclines the width of the waterplane increases, and, consequently, the moment of inertia becomes larger; and since the volume of displacement is constant, the value of B M is increased. From the formula it will be seen that the moment of inertia varies as the *cube* of the breadth, therefore a small increase in breadth can be responsible for largely increasing the moment of inertia, which again causes M to rise. A position can, therefore, in some cases be reached where the increased moment of inertia of waterplane will produce a height of M so that the vertical through the new centre of buoyancy will intersect the vessel's centre line at G, and then a position of rest will be attained, since both the forces of weight and buoyance will be acting on the same vertical line, although in the upright position the vessel was unstable on account of M being below G.

Fig. 28 illustrates this point. In the upright position the centre of buoyancy, metacentre, and centre of gravity are represented by

B, M and G respectively. M being below G, the vessel is, therefore, in an unstable condition. On inclination, the centre of buoyancy travels in the direction shown by B_1 B_2 B_3, etc., and corresponding to these positions we have lines drawn perpendicular to the corresponding water-lines. The moment of inertia being calculated for each water-line, and divided by the volume of displacement, gives the B M, which is set off on these perpendiculars giving M_2 M_3, etc. For the first inclination, which is a small angle, the B M coincides with the upright value, the perpendicular cutting the centre line at M; but on increasing the inclination the altered waterplane

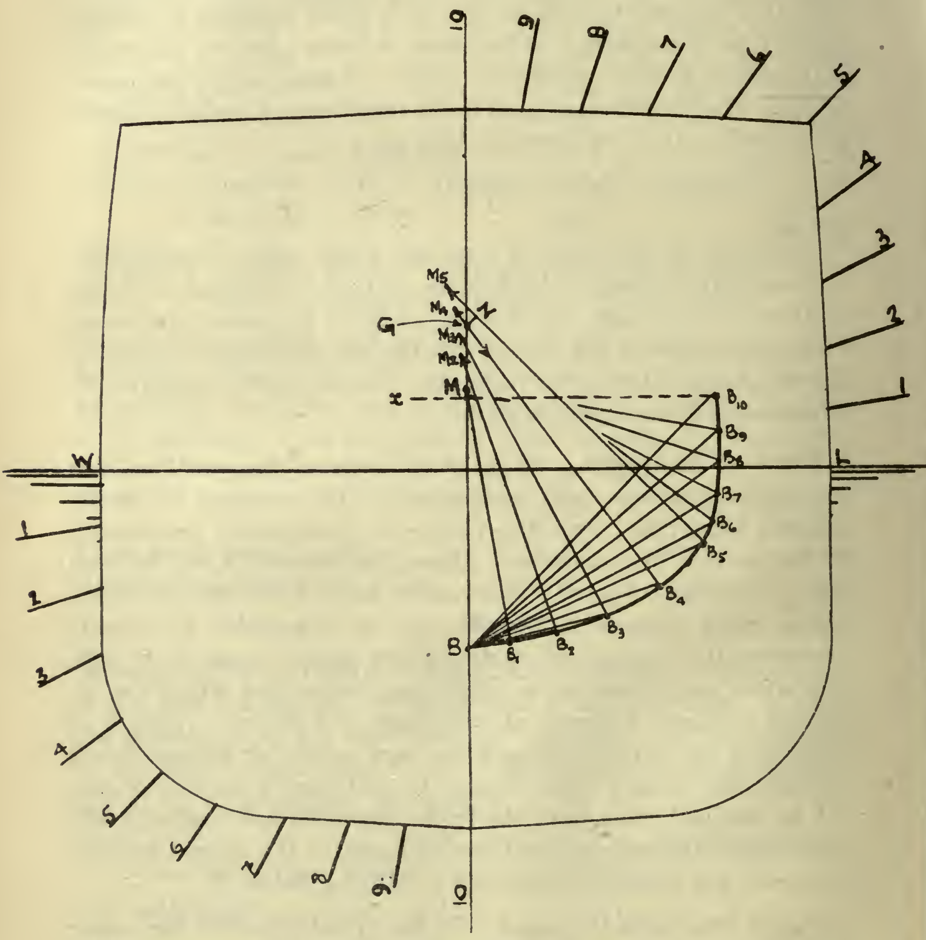


Fig. 28.

width causes a rapid increase in moment of inertia, which now gives larger values for B M, as shown for the succeeding angles. It will be seen that the upward forces of buoyancy no longer intersect the vessel's centre line at M, but at points situated higher. The perpendiculars B_2 M_2 and B_3 M_3 intersect below G, and therefore the vessel is still unstable, and will continue inclining; but on reaching B_4 M_4 this perpendicular is seen to intersect at G, where we now have the forces of weight and buoyancy lying upon the same vertical line, therefore producing equilibrium, this then becoming the position of rest. Now, suppose the vessel to be still further inclined by an external force, such as the wind, so that the centre of buoyancy reaches B_5, the B M here being B^5 M^5, the perpendicular now intersecting the centre line at a point above G, therefore causing a righting lever G Z (the distance between the two forces), which will bring the vessel back to the position corresponding to B_4 G M_4. Of course, it is only in some cases where this occurs; it may happen that successive perpendiculars through the new centres of buoyancy would intersect at points lower than M, instead of increasing in height as described in this case. The above changes are caused by the varying moment of inertia, which, while increasing, gives a large B M and a large shift of the centre of buoyancy, the latter being the means of changing the point of intersection of the buoyancy perpendicular and the vessel's centre line, upon which the centre of gravity lies. B_5 corresponds to an inclination of 45 degrees where the deck-edge enters the water, and at which position the moment of inertia is at its maximum, since the waterplane width is greatest here. The moment of inertia now reduces as the waterplane width becomes less, and the shift of the centre of buoyancy does not increase so rapidly as previously. At 90 degrees of inclination, B_{10} is the corresponding centre of buoyancy, and $B_{10}x$ the line of the upward force, which is here seen to be intersecting below G. If perpendiculars are produced from B_6, B_7, B_8 and B_9, it will be found that the points of intersection with the centre line gradually reduce in height from the position obtained by the perpendicular B_5 M_5.

In the case of a vessel carrying deck loads, such as timber, where the centre of gravity is raised to a high position on account of the cargo on deck, a negative Metacentric Height is sometimes obtained when in the upright position. The result is that the vessel inclines until she reaches an angle where the moment of inertia of waterplane will produce a value of B M so that the vertical through the new

centre of buoyancy will intersect the vessel's centre line at G, and thereby obtain the position of rest as mentioned in the above. The following interesting case of a timber-laden ship was brought to the notice of the author. A certain steamer loading in the Baltic gradually listed as the deck cargo was being put on board. By ballasting she was brought as near to the upright as the capacity of ballast would produce, eventually sailing with a few degrees list to starboard, which during the crossing of the North Sea, increased until the alarming angle of 35 degrees was reached, when an amount of deck cargo on the lower side (starboard) broke adrift, causing the vessel to give a few rolls and to finally settle to a much smaller angle of inclination, but to the port side, which, after trimming some cargo from port to starboard, was further reduced. The causes of the above were as follows : As the bunkers were being burnt out from a low position, the centre of gravity arose from G_1 to G_2 (Fig. 29), and therefore the vessel further inclined so as to obtain stability

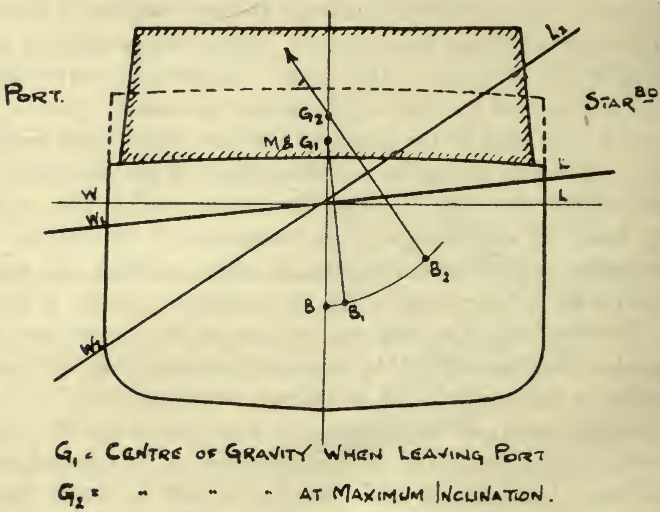


Fig. 29.

in the way mentioned above. When the cargo broke adrift the vessel rolled owing to the sudden withdrawal of weight from the starboard side, and the reason of her taking up the smaller list was because the position of the centre of gravity was now reduced in height, and therefore required a less amount of moment of inertia to obtain the coincidence of the buoyancy perpendicular with that point. An amount of cargo being deducted

from the starboard side, the list was necessarily to port, the trimming of the excess cargo from port to starboard then obviously reducing the list.

Throughout the above it will have been observed that, while a vessel is in her initial upright position, the amount of stability is depending upon the situation of the points **M** and **G**, the latter being fixed and the former varying according to the vessel's underwater form, it being determined by the line of the upward force of buoyancy. M being found by means of the calculations described in the above, it therefore remains to obtain the position of G, the centre of gravity, It must first be found for the light condition, accounting for every item of the hull and machinery by means of the principle of moments. This is an enormous calculation, involving many figures, introducing the possibility of error ; besides, for a large amount of the weight, it is difficult to obtain an accurate position of the centre of gravity. It is, therefore, usual to obtain the position of G by means of an Inclining Experiment performed at the finishing stages of the vessel's construction. For a load condition, the amounts of deadweight are added, and G found as shown in a previous example.

To determine practically, by Inclining Experiment, the height of a vessel's Centre of Gravity: We know that if a weight is moved across the deck of a vessel an alteration in the position of the *centre of gravity* occurs, the C G moving from $\mathbf{G}$ to $\mathbf{G_1}$ (*see* Fig. 30). The distance moved is $\mathbf{G\,G_1}$. We also know that, at all positions of rest, the vessel's *centre of gravity* and *centre of buoyancy* lie in the same vertical line, this vertical being perpendicular to the water-line at which the vessel is floating. This perpendicular cuts the centre line of the vessel's section at a point **M** which, for small transverse inclinations, is the Metacentre, Now, if we know $\mathbf{G\,G_1}$ we can find **G M** (which is the *Metacentric Height*), it being obvious that

$$\mathbf{G\,M} = \frac{\mathbf{G\,G_1}}{\tan\theta} \qquad \theta \text{ being the angle of inclination.}$$

$\mathbf{G\,G_1}$ is readily found by $\dfrac{w \times d}{\mathbf{W}}$, where w = the weight moved across the ship a distance of d, and **W** being the total weight of the ship at the time. To find the angle of inclination that is caused

by the shift of the weight, an Inclining Experiment must be performed as is afterwards described. Having found the value of **G M** by means of $\frac{G\ G_1}{\tan\theta}$ $\left(\text{or } \frac{w \times d}{W \times \tan\theta}\right.$ which is the same thing), we can find the position of **G** relative to the keel by first of all finding the position of **M**, the Transverse Metacentre, as was described in Chapter IV., where we saw that **M** was determined by

$$B\ M = \frac{\text{Transverse Moment of Inertia}}{\text{Volume of Displacement}}$$

Calculating the position of **B** above keel, or scaling it off the curve of vertical centres of buoyancy, usually shown on the displacement scale, and adding this height to **B M**, we have the position of **M** above keel. If from this we subtract **G M**, as found above, we will have the height of *the centre of gravity* of the ship above keel, **G**, of course, lying below **M**, so as to allow of the vessel possessing *stable equilibrium*.

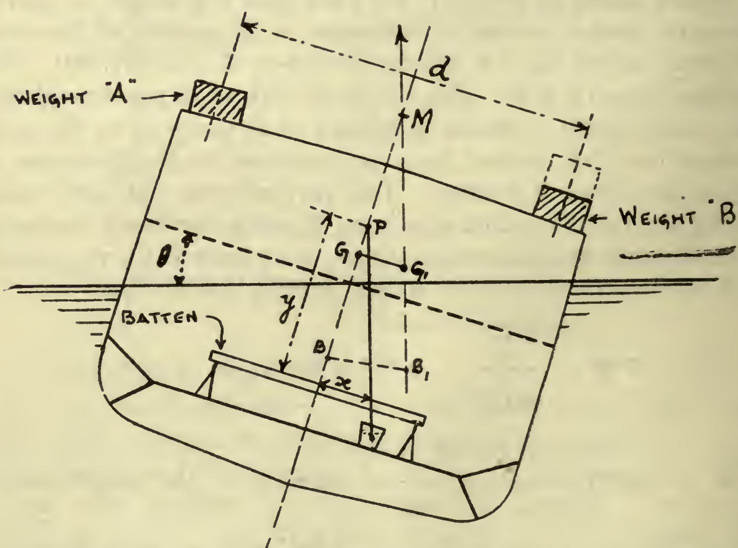


Fig. 30.

Procedure and Precautions.—It will be seen that the following information is required for the Inclining Experiment : The weights that are moved and the distance, the total weight of the ship, the angle of inclination, the height of the *transverse metacentre* at the time (this either being obtained from the *metacentric diagram* or by independent calculation for *moment of inertia of waterplane, volume of displacement and vertical position of the centre of buoyancy*), the weights and positions of all things that have either to go aboard, come ashore or be shifted after the experiment has been performed.

In Fig. 30

G = Centre of gravity in upright position before weight is moved.
$\mathbf{G_1}$ = Centre of gravity in inclined position after weight is moved.
M = The transverse metacentre.
B = The centre of buoyancy in upright position.
$\mathbf{B_1}$ = The centre of buoyancy in inclined position.
P = Point of suspension of the plumb line.
y = Length of plumb line to batten in upright position.
x = Deviation of plumb line.

The batten, upon which the inclinations of the plumb line are marked, must be rigidly fixed. To prevent the plumb line trembling and to help it in becoming steady, before noting the inclination, a bucket of water may be used, as shown in the sketch. Before commencing the experiment a rough estimate of the position of **G** should be made to enable a preliminary calculation to be made to ascertain the probable weight required to give the vessel the angle of heel aimed at. The vessel having been brought to the upright, or as near as can be obtained, a batten is erected upon two trestles at right angles to the plumb line, as shown. The weights that are used should be sufficient to produce a heel of 4 to 5 degrees. The following precautions should be taken : Make a personal inspection of the vessel and ascertain that she is properly afloat ; boilers full to working level ; ballast tanks quite full or empty (it is even better to have them full than to have a small quantity of water rolling from side to side during the experiment) ; fresh water and service tanks full ; that all movable weights are secured ; any boats, floating stages, etc., that are moored to the vessel are removed or slacked off, and all gangways removed. The mooring ropes should be slacked off, and anchors hauled up. No men should be on board other than those actually engaged in the experiment.

The experiment should be performed when there is little or no wind ; if any, the vessel's head should be put to it, if possible.

The weights that are used in the experiment are divided, half being placed on the starboard side and the remainder on the port side, the distance from the centre line being the same in each case. The men employed in the work of moving the weights from side to side must take up a position on the centre line of the vessel when not engaged in the operation.

Everything now being ready, the draughts forward and aft should be taken, so that the actual displacement, transverse moment of inertia, and the position of the centre of buoyancy might be ascertained for the vessel's conditions during the experiment. Before moving the weights the position of the plumb line should be marked on the batten, and the distance from the point of suspension **P** to the batten noted. This is shown in Fig. 30 as y. Now move the weight **A** on to the top of **B**, or in a position so that its distance from the centre line is the same as in the case of **B**. The distance moved is d feet. Note the inclination of the pendulum for future reference by marking it on the batten. The weight **A** should now be moved back to its former position, when the plumb line should correspond with the mark which was placed on the batten before the weight was formerly moved. Should the line and mark not coincide, it proves that there must have been some loose water or something which at the first inclination rolled out of its place and has not been able to regain it. Next move the weight **B** over to a position corresponding with **A**, and note the deviation as before, and which should be the same. Care should be exercised in taking the deviations after the ship has become steady. If there happens to be a slight difference in the deviations for these two shifts (one port and one starboard), the mean should be taken. Generally, two or three plumb lines are used, being placed—one in the forward holds, being allowed to swing in the hatchways ; the second at the fore end of the stokehold or some other suitable position in the engine and boiler space, the trestles being, in both cases, placed upon the tank top ; the third would be placed in the after holds, the trestles being placed upon a lower deck hatch top, so as to be clear of the tunnels. The mean deviation is taken at each plumb line, and the tangent of the angle found by dividing the deviation **x** by the length of the plumb line **y**. This being done for the three

plumb lines, a slight difference may be found in the "tangents," but the mean of the three being found, we may use it in the equation—

$$\frac{w \times d}{W \times \tan \theta} = G\,M$$

Should the vessel be floating with a large amount of trim, it is preferable to make an independent calculation for the displacement, centre of buoyancy and transverse moment of inertia for the draughts at which the vessel floated during the experiment. If the trim is small, these particulars may be taken off the curves, as shown upon the displacement scale, which for ordinary vessels is drawn up for even-keel draughts. From the above it will be seen that the height of the **centre of gravity above the top of keel** =

height of transverse metacentre above keel — G M

$$= M - G\,M$$

$$= (B\,M + h) - G\,M$$

$$= \left(\frac{I}{V} + h\right) - \frac{G\,G_1}{\tan \theta}$$

$$= \left(\frac{I}{V} + h\right) - \frac{w \times d}{W \times \frac{x}{y}}$$

Where **I** = Transverse moment of inertia of waterplane.
V = Volume of displacement.
h = Centre of buoyancy above keel.
W = Total weight of ship plus the weights used in experiment.

The weights and positions of all things that have either to go aboard, come ashore, or be shifted after the performance of the experiment, having been noted, the corrections for them are made as follows: If a weight has to be taken ashore (such as those used in the experiment), multiply the weight by the vertical distance of its centre of gravity from the centre of gravity of the total, as found by the experiment, and divide by the total weight of displacement at the time of the experiment, reduced by the weight of the article that is being taken off, and the result will be the shift of the vessel's

centre of gravity. This shift will be upwards if the weight was removed from a position below G, and downwards had the weight been taken from a position above G.

$$\text{Shift of } \mathbf{C\ G} \text{ vertically} = \frac{w \times d^v}{\mathbf{W} - w}$$

d^v = distance between centre of gravity of weight and vessel's **C G**, including the weight.

If a weight were to be added, multiply the amount by the distance, measured vertically, from its centre of gravity to the vessel's centre of gravity, and divide by the weight of the vessel, at time of experiment, plus the added weight, and the result will be the shift of the centre of gravity of the ship. If the added weight is placed above the former position of G, the shift will be upwards, and if below, it will be in a downward direction.

$$\text{Shift of } \mathbf{C\ G} \text{ vertically} = \frac{w \times d^v}{\mathbf{W} + w}$$

d^v = distance between centre of gravity of weight and vessel's C G. If the weight has only to be shifted to another position in the ship, the correction would be :

Weight multiplied by the distance moved vertically, divided by the displacement $= \dfrac{w \times d^v}{\mathbf{W}}$, G altering in the same direction as the moving of the weight.

Example of inclining experiment performed on steamship 254 ft. × 38 ft. × 18 ft. 8 in. moulded dimensions.

Displacement at time, 1,590 tons.
Weight moved, 25 tons a distance of 30 ft.
Two plumb lines used.
Deviations of forward plumb line, 16 ft. long.

1st shift to	port ...	·93 ft.	·945 ft. mean	·95 ft. mean deviation.
1st ,,	starboard	·96 ft.		
2nd ,,	port ...	·94 ft.	·955 ft. mean	
2nd ,,	starboard	·97 ft.		

$$\frac{\text{Mean deviation, } \cdot95 \text{ ft.}}{\text{Length of plumb line, } 16\cdot0 \text{ ft.}} = \cdot0594 \text{ tangent.}$$

Deviations of after plumb line, 12 ft. long.

1st shift to port ... ·71 ft.	·705 ft. mean	·71 ft. mean deviation.
1st ,, starboard ·70 ft.		
2nd ,, port ... ·72 ft.	·722 ft. mean	
2nd ,, starboard ·725ft.		

$$\frac{\text{Mean deviation, } \cdot71 \text{ ft.}}{\text{Length of plumb line, } 12\cdot0 \text{ ft.}} = \cdot0591 \text{ tangent.}$$

$$\text{Mean of tangents} \left\{ \begin{matrix} \cdot0594 \\ \cdot0591 \end{matrix} \right\} = \cdot05925 \text{ ft.}$$

$$G\,M = \frac{w \times d}{W \times \tan\theta} = \frac{25 \times 30}{1{,}590 \times \cdot05925} = 7\cdot96 \text{ ft.}$$

B M ... = 15·05 ft.

B, above keel = 4·56 ft.

M, above keel = 19·61 ft.

Less G M ... 7·96 ft.

= G above Keel = 11·65 ft. at time of experiment.

Corrections as under:

	Weight Tons.	CG above keel Ft.	Moments.	
Vessel at time of experiment ...	1,590	11·65		18,523·4
Add fittings not yet aboard ...	*15	12·1		181·5
Tons	1,605	—		18,705·0
Deduct—Inclining weights ... 25	305	19·0	475·0	1,022·5
Water ballast ... 265		1·5	397·5	
Staging, tools, rubbish, etc. 15		10·0	150·0	
	1,300 tons.			17,682·5

17,682·5 ÷ 1,300 = 13·6 ft. The corrected height of the centre of gravity above the keel.

(*This total is obtained from a separate calculation taking into account all the various items).

It will be seen that the inclining experiment, while an important thing, is a very simple undertaking. All that is required is to obtain a deviation on the plumb line by listing the ship by means of a known transverse moment obtained, as in the above example, by shifting a weight, or it could be done by the adding of a single weight on one side of the vessel. The whole thing is therefore, within the reach of every ship's officer, provided he is supplied by the builders with the displacement scale and metacentric diagram. For instance, suppose a vessel is about to have a fresh water tank filled, the position of which is at one side of the vessel—say, 20 ft. off the centre line—and the quantity 18 tons. The weather conditions being favourable, no loose water in tanks, mooring ropes slacked off, and everything else in satisfactory condition, a plumb line can be swung and a batten placed in position. Now commence to fill the tank. When full, take note of the deviation of the plumb line. The draughts of the vessel having been taken, the displacement can be read off the scale and the height of the metacentre from the diagram. He would then be in possession of everything necessary to give him the amount of metacentric height at the time, or the height of the vessel's centre of gravity above the keel. Say displacement, including the weight of the added fresh water, was 3,600 tons, and M 18 ft. above keel, the deviation obtained with a 20 ft. plumb line being ·5 ft. The inclination obtained by filling the tank at a distance of 20 ft. off the centre line is obviously equal to moving the same amount of weight that distance across the deck, since had it been placed upon the centre line no inclination would have resulted, therefore, the added weight of 18 tons is multiplied by 20 ft. to obtain the inclining moment.

$$\text{G M} = \frac{w \times d}{\text{W} \times \dfrac{\text{deviation}}{\text{length of plumb line}}} = \frac{18 \times 20}{3{,}600 \times \frac{\cdot 5}{20}} = 4 \text{ ft.}$$

$$\text{M} = 18 \text{ ft. above keel}$$
$$\text{G M} = 4 \text{ ft. (as found by experiment)}$$

therefore $\quad \text{G} = 14 \text{ ft. above keel.}$

CHAPTER VI.

Co-efficients for Heights of Centre of Gravity in Typical Vessels. Effect of Free Water in the Tanks, or Liquid Cargo in the Holds of Vessels. Effect on Initial Stability due to Adding Water Ballast in Double-Bottom, Deep Tanks and on Deck, etc.

Co-efficients for Heights of Centre of Gravity.—In the designing stages the height of the centre of gravity is generally required, and is found either by the lengthy detailed calculation comprising every item of the hull and machinery, or by the quicker method of comparison with a previous similar ship whose height of G, as obtained by inclining experiment, is known. The method adopted for the latter is to obtain the height of G for the previous ship, as a proportion or co-efficient of the depth, moulded. Say, a vessel of 20 ft. depth, moulded, has her centre of gravity at 12 ft. above the base line (*i.e.*, the top of keel), the height of G would be :

$$\frac{12}{20} = \cdot 6 \text{ of the depth, moulded.}$$

Say, the new vessel has a depth, moulded, of 22 ft., then 22 × ·6 = 13·2 ft., which would be very near to the height of G for this vessel, assuming her to be exactly similar in arrangement and distribution of weight. Should there be differences, they should be corrected for ; for instance, suppose the above new vessel has an addition in the shape of a longer bridgehouse, which is situated at 26 ft. above the keel, the extra weight being 10 tons and the total weight of the new vessel 1,520 tons, then—

	Weight.		Lever.	Moment.
Vessel exactly similar to the previous ship, *i.e.*, without the amended bridge...	1,510 tons	×	13·2	19,932
Addition in arrangement over the basis ship	10 ,,	×	26·0	260
Total weight of new ship ...	1,520 tons.			20,192

20,192 ÷ 1,520 = 13·28 ft., which is the corrected height of G for the new ship.

Of course, this is a very simple case, with only the one amendment, but it will suffice as an example. It is very probable that when comparing a proposed ship with a previous vessel a fairly large amount of corrections will be necessary, but, if carefully made, the result obtained by use of this method will be very nearly correct, and in some cases may be even more accurate than a figure obtained by means of the independent calculation. The following are average figures for the Co-efficient for Height of Centre of Gravity of hull and machinery complete :

Type.	Co-efficient.	Depth, Moulded, taken—
Fast Atlantic Liner with shelter deck plus large erections above	·60	to shelter deck.
Cargo and Passenger, about 16 knots with shelter deck and erections above	·57	to shelter deck.
Cargo Vessel with shelter deck	·54	to shelter deck.
Cargo vessel with poop, bridge and forecastle covering about half length	·65	to upper deck.
Fast Channel Steamer with awning deck ...	·60	to awning deck.
Coasting Steamer with long raised quarter deck, bridge and forecastle, also double-bottom	·72	to a mean between main and r.q. decks.
Steam Trawler	·75	to main deck.
Sailing Vessel	·70	to upper deck.

Effect of Free Water in the Tanks, or Liquid Cargo in the Holds of Vessels. A great loss of stability is caused by a free surface of water, etc., in tanks or holds of vessels. The free surface of a vessel's cargo, such as oil or grain, has a most dangerous effect if not guarded against, since if an external force inclines the ship, the cargo will move in the inclined direction and, of course, further increase the amount of inclination due to the applied force. When such cargoes are carried, every precaution should be taken to prevent a free surface by keeping the holds quite full and to erect longitudinal divisional bulkheads, so that if a free surface does occur, the movement will not be large. A loose way of speaking is to say that a free surface raises the centre of gravity; what is meant is that it "virtually" raises the centre of gravity. Take the case of a vessel with a free surface of water in a double-bottom tank. Fig. 31 represents a vessel with a double-bottom compartment partly filled and heeled over to a small angle, though exaggerated in the sketch for the sake of clearness.

$w\ l$ = Free water surface in the tank, and is parallel to WL, the water-line of the vessel before inclination.

$w_1\ l_1$ = The new surface of water in the tank, and is parallel to $W_1\ L_1$, the water-line of the vessel after inclination.

Let v_f = the volume of water contained in either of the wedges in the tank, $w\ A\ w_1$ or $l_1\ A\ l$,

g and g_1 = their centres of gravity,

b = the centre of gravity of the water before inclination.

b_1 = ,, ,, ,, after ,,

V_f = the total volume of water in tank,

$$\text{therefore } \frac{v_f \times g\ g_1}{V_f} = b\ b_1$$

m = the point of intersection of a perpendicular through b_1 and the centre line of the vessel upon which lies the position of b.

$b\ m$ is obviously equal to $\dfrac{b\ b_1}{\tan\ \theta}$ and since $b\ b_1 = \dfrac{v_f \times g\ g_1}{V_f}$

$$\text{we have } b\ m = \frac{v_f \times g\ g_1}{V_f \times tan\ \theta}$$

or $b\ m = \dfrac{i}{V_f}$ where i = the moment of inertia of free water surface about the fore and aft axis passing through A, and V_f the volume of water in tank, the proof

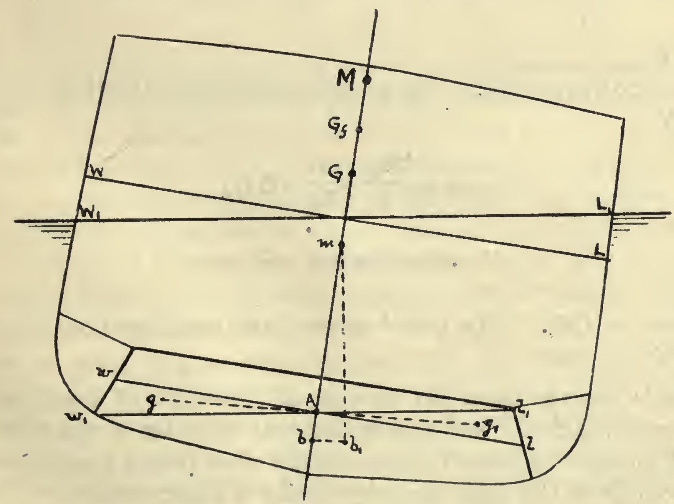


Fig. 31.

of this latter equation is the same as that for B M previously dealt with. Although b is the "actual" centre of gravity of the water in the tank, its effect upon the ship is as though it were at m, which is termed the "virtual" centre of gravity of water in tank. It is like the case of a pendulum where the centre of gravity is at the point of suspension, which in this case is m. In the sketch **G** represents the position of the centre of gravity of the vessel, including the water, while in the upright position: but on the slightest inclination the centre of gravity shifts to G_f, because of the centre of gravity of the water being virtually shifted to m. The effect is similar to shifting the amount of water in tank a vertical distance of $b\,m$. The value of the shift GG_f, which is the loss of metacentric height, is therefore equal to

$$\frac{W_f \times b\,m}{W}$$ where W_f = the weight of the water in tank (tons) and W = the total displacement of vessel, including water in tank (tons).

Converting these weights into volumes we have—

$$\frac{V_f \times b\,m}{W \times 35} \quad GG_f$$ (W being multiplied by 35 for volume),

$$= \frac{V_f \times b\,m}{V} = GG_f,$$ or it may be written

$$\frac{V_f}{V} \times b\,m = GG_f.$$ But we have already seen that $b\,m = \frac{i}{V_f}$,

$$\text{therefore } \frac{\cancel{V_f}}{V} \times \frac{i}{\cancel{V_f}} = GG_f.$$

V_f cancelling out, we have—

$$\frac{i}{V} = GG_f.$$ We therefore see that the alteration GG_f depends entirely upon the amount of "moment of inertia of the free water surface" and not the amount of water in the tank. A small quantity of water may have more effect than a large quantity, on account of the small quantity having a larger surface. It being difficult to obtain the correct moment of inertia of the free water

surface, and since such is necessary so as to obtain the true position of m, this method is not quite satisfactory in the case of making a correction to the result of an inclining experiment. A better and a more reliable method is to take the two wedges $w\ A\ w_1$ and $l\ A\ l_1$ and treat them as is done with the wedges of immersion and emersion in an ordinary case for finding the shift of a vessel's centre of buoyancy—that is, to find the moment of transference of the wedges and adding it on to the inclining moment caused by the moving of the weights across the deck, thereby obtaining the total moment which produced the angle of inclination. The addition to the moment will, therefore, be the weight of a wedge multiplied by $g\ g_1$,

the weight of a wedge being equal to $\dfrac{v_f}{35 \text{ for salt water}}$ Therefore,

$\dfrac{v_f}{35} \times g\ g_1 =$ the addition to be made to the moment caused by shifting the weights. G M at the time of experiment will, therefore, equal—

$$\frac{w \times d + \left(\dfrac{v_f}{35} \times g\ g_1\right)}{W \times \tan\theta}$$

If the double-bottom were divided by a longitudinal centre water-tight girder, as shown in Fig. 32, the inclined water surface would take up the positions as shown by $w_2\ l_2$ and $w_3\ l_3$. In this case we have a wedge $w\ a_2\ w_2$ being shifted through a distance of $g_2\ g_2$ to $l_2\ a_2$ A, and another similar wedge, upon the opposite side of the girder, being shifted a similar distance. It will be seen that the volume of either of the transferred wedges is only one-quarter of the volume of a transferred wedge when there was no division in the tank, and that the transverse shift $g_2\ g_2$ or $g_3\ g_3$ is only one-half as great as $g\ g_1$, therefore the shift of a single wedge causes a moment which is only $(\frac{1}{4} \times \frac{1}{2}) = \frac{1}{8}$th of the amount caused by shifting a wedge when the tank had no centre division. But we now have a shift of two wedges occurring, one on each side of the centre line, therefore the shift of the two smaller wedges will cause a moment equal to $\frac{1}{8} \times 2 = \frac{1}{4}$ of that caused when the tank was

not divided, and when only one large wedge was transferred. In the above it is supposed that the vessel is prismatic (being of the same section throughout), but, roughly speaking, we could say that in the case of ordinary vessels the effect of free water in a vessel with a water-tight centre girder is only one-quarter of that in a vessel without same. In the case of a vessel where the centre girder is not water-tight, but the holes through same are only small such as drainage holes, it is practically impossible to estimate the effect, as the levels of the water on each side may be at some intermediate position to those shown in Figs. 31 and 32 at the time of her beginning to roll back to the upright, since it will not have had time to flow through the small holes and obtain the same level;

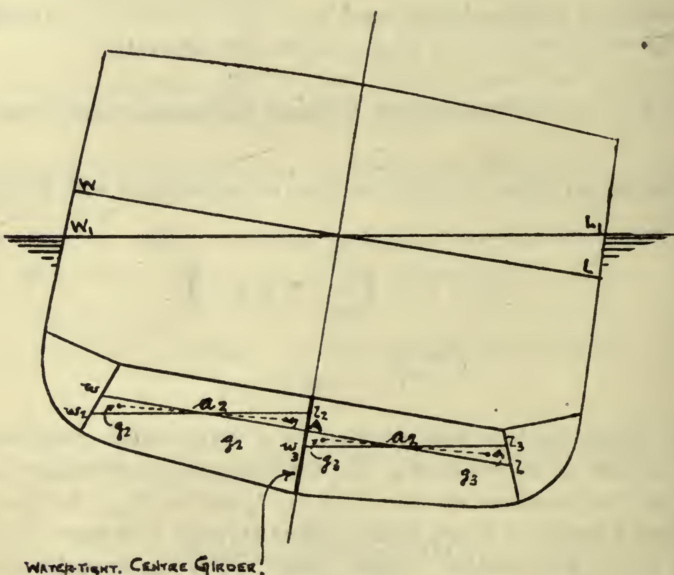


Fig. 32.

but in the case of an inclining experiment, when the vessel is held over to an angle for some time, the water has time to run through and find its level on the opposite side. From this it will be seen that great advantage is obtained by cutting nothing more than small drainage holes through the centre girder, and a still greater advantage by making it water-tight. While "corrections" for free water may be made as above described, yet the description goes to prove the great necessity for having no loose water in a vessel

at the time of an inclining experiment, it being difficult to estimate the required amount of "correction" to the degree of accuracy necessary when making corrections to the result obtained from that operation. The above also applies to cargoes with a free surface in the holds of vessels; the effect on the vessel's stability due to the erection of longitudinal bulkheads or shifting boards being exactly similar to the case of fitting the water-tight girder in the above ballast tank.

It has been shown that moment of inertia of the surface of free water, divided by the volume of the vessel's displacement, equals the loss in metacentric height caused by the free water surface.

$\dfrac{i}{V}$ = loss if the liquid is water; but if it is other than water, and the specific gravity different to that of the water in which the vessel is floating, then GG_f, or the loss of

$$G\,M = \frac{i \times \text{weight per cubic foot of the liquid, in tons.}}{\text{displacement in tons.}}$$

Effect on Initial Stability due to adding Water Ballast.—The present-day means of ballasting ships is to build tanks into the structure, into which water is allowed to enter when extra immersion is required or for the purpose of providing stability. The most usual positions for water ballast are shown in Fig. 33. The tank marked 1 is the "double-bottom" tank, being shown all fore and aft in the hold space, and is also seen in sections **a**, **b**, and **c**; 2 is

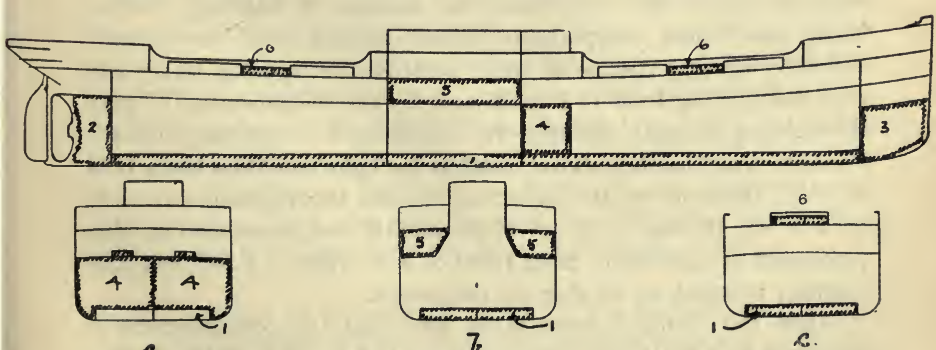


Fig. 33.

the after peak tank, and 3 the fore peak tank ; 4 is a "deep" tank divided longitudinally by a centre line water-tight bulkhead, also shown in section **a** ; 5 are 'tween deck side tanks, being shown in section by **b**. Another means of ballasting which, in some cases, has been fitted, is to build small tanks upon deck between the hatchways, etc. ; these are shown in the elevation, and also in section **c** by 6. The filling of the various tanks has important effects on the vessel's stability due to altering the position of the centre of gravity, and also the immersion ; the alteration in the immersion produces a new position of the transverse metacentre, since the moment of inertia of waterplane and the volume of displacement are changed, while the centre of gravity changes its position by moving in the direction of the added ballast. Both of these alterations must be taken into account when determining the change in stability, and for this a metacentric diagram is of much use. Taking the above-mentioned tanks, we may note their resultant effects upon a vessel whose curve of transverse metacentres is shown in Fig. 34. At the light draught, 7 ft., the position of **G** is shown to be 20 ft. above the base, the **GM**, therefore, being 5 ft., since the curve of metacentres gives the height of **M** as 25 ft. for this draught. Now, suppose the double-bottom tanks, the capacity of which is 400 tons, to be filled, the effect being to reduce the position of **G** to 15 ft. above the base, and to increase the draught to 10 ft. At this new draught **M** is now found to be 22 ft. above the base, the curve at the lower draughts descending and thereby causing this alteration. **G M** is now 7 ft., which shows that the vessel possesses a larger amount of G M in the new condition. The metacentric height alone is not, however, the true means of comparing the stability of the two conditions, the amount of righting moment being the truest comparison. When dealing with metacentric stability, we saw that G M sin θ = G Z, the righting lever, and that the righting lever G Z multiplied by the displacement W gave the righting moment, therefore W $\times$ G M sin θ = righting moment. We have seen that the above vessel in the light condition has a G M of 5 ft., therefore at 10 deg. of inclination the righting lever will be 5 $\times$ sin 10 deg. = 5 $\times$ ·1736, and if the corresponding displacement is 1,500 tons, then 1,500 $\times$ 5 $\times$ ·1736 = 1,302 foot-tons righting moment at 10 deg. of inclination.

When the double-bottom tanks are filled, the displacement is increased to 1,900 tons, and we have already seen that the G M is then 7 ft. The righting moment for the ballasted condition at 10

deg. of inclination is therefore $1900 \times 7 \times \cdot 1736 = 2{,}309$ foot-tons. The righting moment for the ballasted condition is seen to be about 77 per cent. more than when light, the difference being caused by an increase of displacement and an increased metacentric height. Comparing the metacentric heights only, the increase is 40 per cent. The vessel will be much stiffer in the ballasted condition, and will most probably roll heavily. This difference in stability between the light and ballasted conditions is very representative of the modern cargo tramp of full form, the ballasting in this manner of such vessels in many cases having a similar effect, as shown in this example, and is responsible for the uncomfortable rolling and the heavy strains obtained in these vessels when ballasted and in a seaway. By filling the aft and fore peak tanks, Nos. 2 and 3 in Fig. 33, the change in the vertical position of the centre of gravity is usually of small amount, although they may effect the immersion to a fair extent. They are each extremely useful for trimming purposes, on account of the large "moment" produced by their being placed at the extremities of the vessel. When a vessel is without cargo it is very necessary that, apart from the necessity of the provision of stability when such is required, ballast should be provided so as to obtain greater immersion for the propeller and thereby obtain greater efficiency, and save time when in a seaway as well as to minimise the risks of the breaking of the tail shaft due to racing when the vessel is pitching and heaving; the increase of draught also making the vessel much easier to navigate under such circumstances. As in the above example, the latter necessities are often provided at the expense of certain disadvantages. Of course, it would be possible to obtain the required amount of immersion for the propeller by placing a sufficient quantity of ballast in the after peak. In such a manner the increase of immersion aft could be obtained without making the vessel excessively "stiff" as is often the case in the above-mentioned type, when the ballast is fitted to a low position all fore and aft, but the trimming of the vessel would lift her higher at the fore end, and possibly be disadvantageous from the point of view of navigation, since with a beam wind the tendency is for the vessel's head to fall off.

The position of M being governed by the underwater form and shape of the vessel, it will be seen that the only means of controlling the stability in such a case is by the position of G. In some modern vessels of large beam the excess of stiffness prevails to a large extent,

but in many cases of this type the disposition of ballast is now arranged so that the position of G produced by filling the tanks is such as will give a metacentric height which will result in easy rolling when in a seaway as well as a reasonable amount of stability. In such cases additional ballast tanks are fitted, which not only have the effect of further increasing the immersion, but to raise the centre of gravity to a higher position than would be obtained in the usual way by filling the double-bottom and peak tanks only. The additional ballast is placed in " deep " " 'tween deck side," or " deck " tanks, as previously mentioned and shown in Fig. 33. Deep tanks are usually situated near 'midships and are used as holds when the vessel is in a " load " condition. Side tanks in the position as shown by Fig. 33 may be used as bunkers when required. While the effect of " deep " tanks is to reduce the excess of " stiffness " when ballasted, yet in many cases where they are fitted this is still found to exist. Such tanks also possess the great disadvantage of setting up serious strains, owing to a large weight being confined to one particular point. In the case of the patent centilever-framed type of ship built by Sir Raylton Dixon and Co., Limited, both the above disadvantages are done away with. The additional ballast being placed in compartments extending all fore and aft below the deck and next to the vessel's side, has the effect of raising the centre of gravity, and thereby preventing the excessive " stiffness " and heavy rolling, while the distribution is such as will cause no strain to the vessel's structure. A large number of vessels of this type have been built, giving excellent results. When the " double-bottom " and the additional tanks are filled, the immersion is such as will allow of easy and safer navigation when without cargo in bad weather, the position of the additional ballast producing ideal stability conditions.

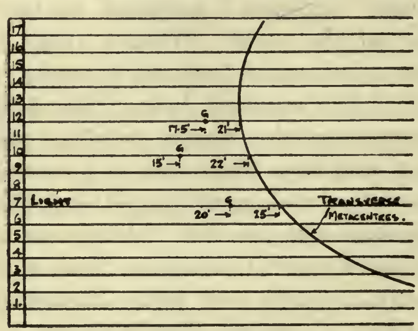


Fig. 34.

Suppose that by filling the aft peak, fore peak and deep tanks, the above vessel's draught is increased to 12 ft., the displacement there being 2,200 tons, and the centre of gravity now being 17·5 ft.

above the base. The curve of metacentres here shows the metacentre to be 21 ft. above the base, giving a G M of 3·5 ft. For this condition at 10 deg. of inclination the righting moment $= 2{,}200 \times 3{\cdot}5 \times {\cdot}1736 = 1{,}336{\cdot}7$ foot-tons, which shows the vessel to have a righting moment which is very little in excess of the amount when in the absolutely "light" condition.

The following is an interesting point in connection with the ballasting of a certain vessel known to the author. It is generally the case that when a double-bottom compartment is completely filled, the value of G M has been increased; the fact of the addition of ballast at such a low position giving one, at first sight, the impression that G has been lowered and the G M thereby increased. This case shows how this might not always happen, the G M being less when in the ballasted condition than when light. Fig. 35 is the metacentric diagram for the vessel in question. Before filling the tanks the draught is *a*, and after filling it is *b*. It will be noticed that by the addition of the ballast we have lowered the centre of gravity from G to G_1, which fact, at first sight, makes one think that G M is now larger. It will be seen that in this instance such is not the case, because by adding the extra weight we have immersed the vessel to a deeper water-line, where M is much lower than it was when at the light draught, the reduction in the height of M being greater than in the case of G. In ordinarily proportioned cargo vessels of full form, M is usually decreasing rapidly about the light draught, in this case the decrease being exceptionally rapid, and the amount being more than the decrease in height of G, therefore, at *b* draught the metacentric height G_1 M_1 is smaller than G M at *a* draught. However, in such a case, notwithstanding the reduction in G M, it may happen that owing to the increase of displacement, the righting moment at *b* draught may be greater than when at *a* draught.

In the curves of metacentres for ordinary vessels, and especially when the lines are of full form, it will be observed, as in Figs. 34 and 35, that M is highest at the light draughts at which the curve is rapidly descending. Near to the load draught it is seen to be again ascending. The reason for this variation is as follows: At the light draughts of such vessels the waterplane is fairly large and wide, producing a proportionate transverse moment of inertia, and the volume of displacement being small, the value of B M is large since

$$\frac{\mathrm{I}}{\mathrm{V}} = \mathrm{B\ M}.$$

As the draughts increase, the height of M is lowering owing to the volume of displacement increasing at a greater rate than the moment of inertia of waterplane, since the fulness of the waterplanes is increasing slowly. although the height of B is increasing and consequently tending to lift M, yet the effect due to the moment of inertia and volume of displacement is predominant. When the deeper draughts are reached, the displacement does not increase so rapidly, because the vessel is of a wall-sided shape over a large portion of her length at these draughts, and, therefore, the rate of increase of moment of inertia and volume of displacement becoming nearly equal, the value of B M is now decreasing so slowly that the increasing height of B has the effect again to raise M, since B is increasing in height at a quicker rate than B M is lessening at these draughts.

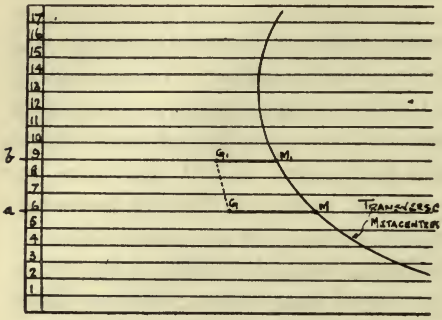


Fig. 35.

CHAPTER VII.

EFFECT ON STABILITY WHEN PASSING FROM SALT TO FRESH WATER, OR VICE-VERSA. LOSS OF INITIAL STABILITY DUE TO GROUNDING; THE CASE OF A VESSEL IN DRY DOCK. EFFECT ON STABILITY DUE TO VESSEL BEING PARTLY IN MUD.

Effect on passing from Salt to Fresh Water or vice-versa. The draught being increased when a vessel passes from salt into fresh water, there will be a new position of M, because the moment of inertia, volume of displacement and the vertical centre of buoyancy will have altered. The position of G, of course, does not alter. In the diagram shown by Fig. 36 it will be seen that the curve of metacentres descends until a certain draught is reached, and then it again ascends. The lowest point is l, and is at the draught **A**. If the draught in salt water is less than **A**, say **B**, then the metacentre will be lowered when passing into fresh water, on account of the vessel sinking deeper. If the draught in salt water is greater than **A**, say **C**, then the metacentre becomes higher when the ship passes into fresh water, because the curve is here ascending. We therefore see that the stability alters when passing from sea to fresh water in a manner which is governed according to the position of the original water-line. The above conditions are reversed when passing from fresh to salt water, Passing from salt to fresh water, and increasing the draught, would give less freeboard, and would slightly reduce the range of stability. The effect of freeboard is dealt with in the following chapter in connection with large angles of inclination.

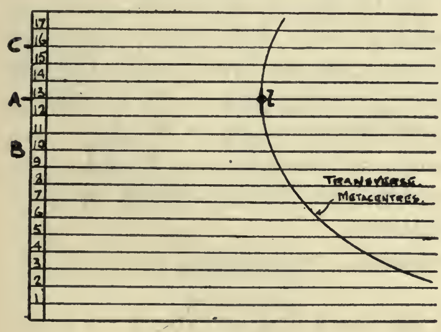


Fig. 36.

Loss of Initial Stability due to Grounding. When a vessel first takes the ground—that is, *just* touches it—the stability is not affected so long as the tide remains as high as the water-line that

she previously floated at, because she will still displace the same amount of water as before, and thereby still receive the same upward support; but suppose the tide to fall, we know that the vessel will then lose a layer of buoyancy contained between the old and new water-lines. She will now obtain an amount of support from the ground, which will act vertically upon the keel if the upright position is preserved, and this amount of support must be equal to the lost layer of buoyancy. When the vessel is inclined to a small angle, so that the bilge does not touch the ground, this new upward force actually tends to upset the vessel.

In considering the question of this particular loss of stability, a good way is to take the case of a vessel in dry dock, just after she has taken the blocks, and without any shores in position, as illustrated by Figs. 37 and 38. In these sketches—

W L = The water-line when the vessel is afloat and upright.

W_1 L_1 = The water-line when the water has receded and left the vessel upright upon the blocks, as in Fig. 37.

W_2 L_2 = The water-line when the water has receded, but the vessel now being inclined upon the blocks, as in Fig. 38.

B = The centre of buoyancy corresponding to **W L** (afloat).

B_1 = The centre of buoyancy corresponding to **W_1 L_1**.

B_2 = The centre of buoyancy corresponding to **W_2 L_2**. (The centre of buoyancy now having shifted outward owing to the inclination).

M = The transverse metacentre corresponding to water-line **W_2 L_2**.

W = The total weight of the vessel and everything on board being equal to the displacement of water when afloat.

G = The centre of gravity of the vessel's weight **W**.

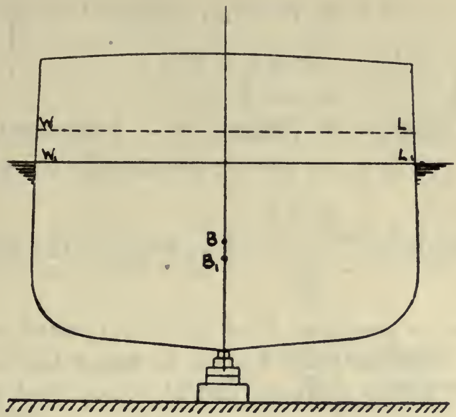
P = The weight taken by the blocks, which is obviously equal to the amount of the displacement contained in the layer between **W_1 L_1** and **W L**, since this amount has been lost owing to the lowering of the water-line.

W – P = The buoyant support of water acting vertically upwards through **B_2**, being equal to the total displacement minus the lost layer.

O = The position of the "effective metacentre" after the vessel has taken the blocks.

L = The point of contact of keel and blocks.

Fig. 37.



The angle of inclination is supposed to be small, although it is shown exaggerated in Fig. 38 for clearness. On looking into Fig. 38 it will be seen that we have one downward force **W** acting through the vessel's centre of gravity **G**, and two upward forces, **P** acting through **L**, the point of contact with the blocks; and **W–P** acting through $\mathbf{B}_2$, the centre of buoyancy of the displacement in the inclined position. If we can find the point **O**—where the resultant upward force of **P** and **W–P** cuts the centre line of the vessel—we will have the position of the "effective metacentre." The resultant of these two upward forces is shown by the dotted line which passes through **O**, the amount of this combined force being exactly equal to the downward force **W** acting through **G**. From the following it will be seen how this resultant is found by the combination of the two upward forces. Using the principle of moments, and taking "levers" from the line of force **W–P**, we have "moments" about that line as follows:

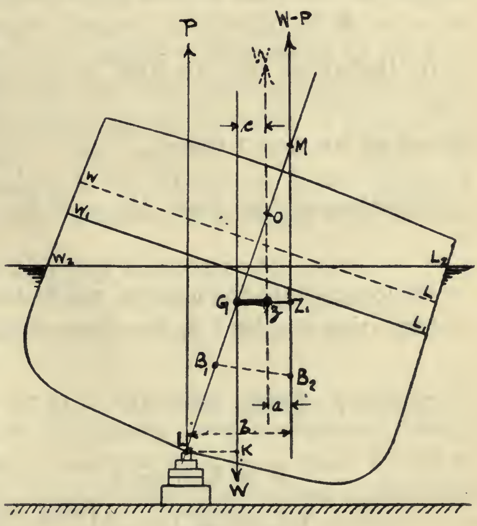


Fig. 38.

$$\begin{array}{rcl} & \text{W–P} \times \text{nil} = \text{nil.} & \\ & \text{P} \times b = \text{P} \times b & \\ \hline \text{Totals} = & \text{W (weight)} \quad \text{P} \times b \text{ (moment)} & \\ \hline \end{array}$$

therefore $\dfrac{\text{P} \times b}{\text{W}} = a$, which is the distance from **W–P** that a resultant *W* of the two upward forces will act. We have now only two lines of forces to deal with, W acting downwards, and **W** acting upwards, and if we can find the distance **c** between them, we will have the "righting" or "upsetting lever," as the case may be (in this case we have a "righting lever.") If we suppose the vessel to be floating at $\mathbf{W_2}$ $\mathbf{L_2}$, the righting lever would be $\mathbf{GZ_1}$, the distance between the downward force of gravity and the upward force of buoyancy; but on account of having to contend with the upward force **P** from the blocks, we have seen that the resultant of the two upward forces is at a distance *a* inside of the vertical W–P, and consequently the new effective righting lever will be shorter than $\mathbf{GZ_1}$ by the amount *a*.

Effective righting lever, $\text{G}z = \text{G Z}_1 - a$.

In the above we saw that $a = \dfrac{\text{P} \times b}{\text{W}}$

therefore we may write—

Effective righting lever $\text{G}z = \text{G Z}_1 - \dfrac{\text{P} \times b}{\text{W}}$

On looking into the figure it will be seen that $b = \text{L M} \sin \theta$, and substituting this for b in the above equation we have—

Effective righting lever $\text{G}z = \text{G Z}_1 - \dfrac{\text{P} \times \text{L M} \sin \theta}{\text{W}}$

$$\left(\frac{\text{P} \times \text{L M} \sin \theta}{\text{W}} \text{ being equal to } a\right)$$

Loss in metacentric height $= (\text{G M} - \text{G O}) = \text{O M}$

$$= \left(\frac{\text{G Z}_1}{\sin\theta} - \frac{\text{G}z}{\sin\theta}\right) = \frac{a}{\sin\theta}$$

since the difference between G Z_1 and Gz is a.

Above we have seen that $a = \dfrac{\text{P} \times \text{L M} \sin\theta}{\text{W}}$, so therefore the loss in metacentric height $= \dfrac{a}{\sin\theta} = \dfrac{\text{P} \times \text{L M} \cancel{\sin\theta}}{\text{W} \times \cancel{\sin\theta}}$ sin θ cancelling out we have

Loss in metacentric height $= \dfrac{\text{P} \times \text{L M}}{\text{W}}$, (which is best for ordinary use in finding the loss in metacentric height), and, when deducted from the original amount, readily gives the state of the vessel's stability for any required condition upon the blocks. This formula being simple, is easily applicable. **P** is quickly found from the displacement scale, being the difference between the displacement of water when afloat and when upon the blocks. **L M** is obtained by reading off the metacentric diagram, the height of **M** above the base for the new water-line, and adding to this depth of the keel. **W** being the total displacement, has been already found from the displacement scale when obtaining the displacement of water when afloat. Of course, throughout the question we have made the assumption that the inclination is one within the limits for which the position of M is practically constant, and it should be noted that

$$\text{B}_1\ \text{M} = \frac{\text{Moment of inertia of the new waterplane.}}{\text{Volume of displacement up to new waterplane.}}$$

Effect on Stability due to a Vessel being partly in mud. In Fig. 39 we have shown a vessel floating freely in water of density equal to, say, 1, $\text{W}_0\ \text{L}_0$ being water-line and B_0 vertical centre of buoyancy, the vessel's keel just touching the level of a mud line of specific density of, say, S_1. Draught to $\text{W}_0\ \text{L}_0 = \text{D}_0$.

In Fig. 39a the water-line W_0 L_0 has fallen to W_1 L_1 and vessel has become partly submerged in mud to water-line W_2 L_2. The vessel's draughts to W_1 L_1 now equal, say, D_1 while the vertical centre of buoyancy equivalent to W_1 L_1 is B_1. However, it is quite obvious that the greater density of the mud portion up W_2 L_2 acting at its own vertical centre of buoyancy B_2 (*See* Fig. 39b), will tend to lower B_1 to, say, B_3.

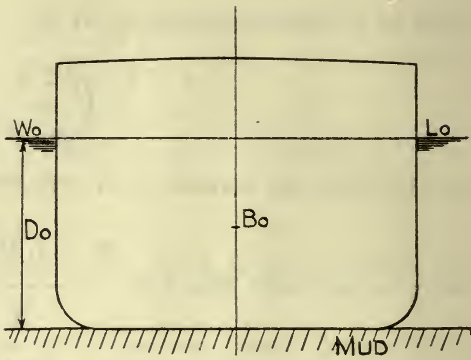


Fig. 39.

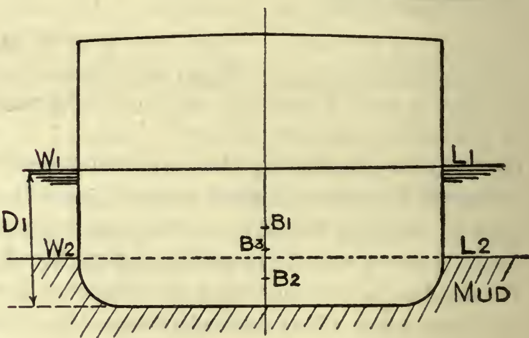


Fig. 39a.

It is desired to deduce a formulæ to use in obtaining the depth vessel is submerged in the mud, *i.e.*, D_2, Fig. 39b. We intend to treat the additional buoyancy below W_2 L_2 as an increment of displacement over and above the displacement up to W_1 L_1.

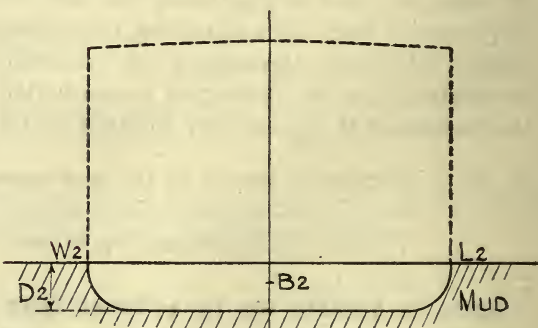


Fig. 39b.

That is if W_0 be displacement up to W_0 L_0 and W_1 be displacement up to W_1 L_1 and W_2 $(S_1 - 1)$ is additional displacement up to

$W_2\ L_2$, then $W_o = W_1$ plus $W_2\ (S_1 - 1)$, and assuming length of vessel equal L and breadth = B.

and C_o = displacement co-eff. up to $W_o\ L_o$.
C_1 = ,, ,, ,, ,, $W_1\ L_1$.
C_2 = ,, ,, ,, ,, $W_2\ L_2$.

$$\text{Then } \frac{\cancel{L} \times \cancel{B} \times D_o \times C_o}{\cancel{35}} = \frac{\cancel{L} \times \cancel{B} \times D_1 \times C_1}{\cancel{35}} + \frac{\cancel{L} \times \cancel{B} \times D_2 \times C_2\ (S_1 - 1)}{\cancel{35}}$$

or 35, L and B cancelling out—

$$D_o \times C_o = D_1 \times C_1 + D_2 \times C_2\ (S_1 - 1)$$

$$D_2 = \frac{D_o \times C_o - D_1 \times C_1}{C_2\ (S_1 - 1)}$$

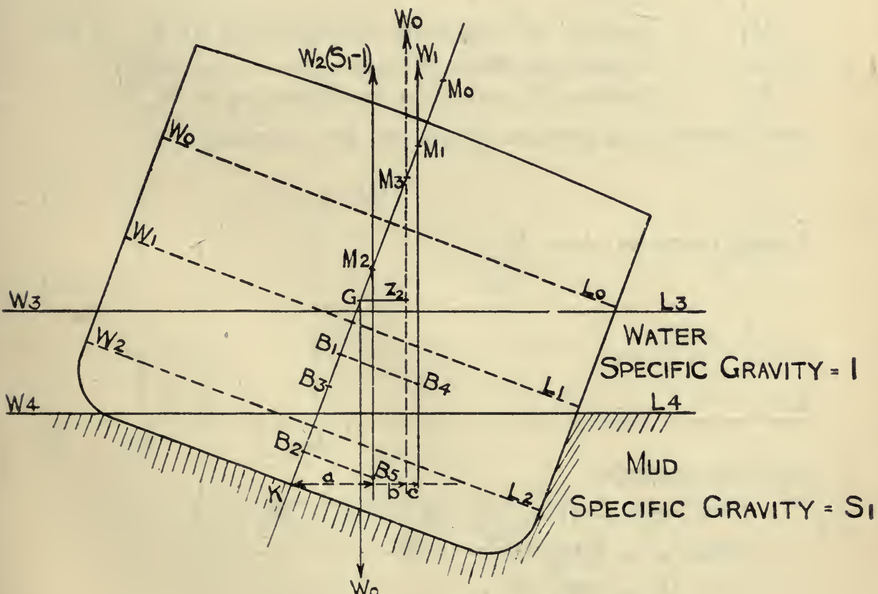


Fig. 39c.

or if vessel of prismatic form—

$$D_2 = \frac{D_o - D_I}{S_I - l_I}$$

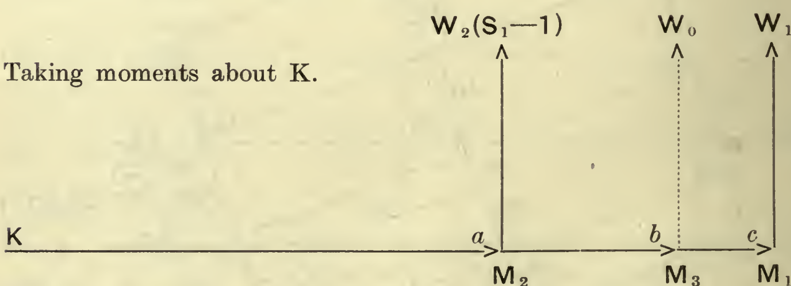
It is now worth while considering what effect this condition may have on the transverse stability. This will involve our obtaining new metacentric height, new righting lever and new righting moment supposing the vessel is inclined to a small angle of heel θ, and that the mud is of even density.

In Fig. 39c we have exaggerated the angle of heel for clearness. Then in Fig. 39c.

$W_3 L_3$ = inclined water-line corresponding to W_I L_I.
$W_4 L_4$ = ,, mud line ,, ,, W_2 L_2.
B_4 = ,, centre of buoyancy of vessel up to W_3 L_3 (neglecting lowering effect of mud).
B_5 = ,, centre of buoyancy up to W_4 L_4.
G = position of centre of gravity of vessel.
M_o = position of metacentre corresponding to original water-line W L when vessel floating freely in water only.
M_I = position of metacentre corresponding to W_I L_I and neglecting effect of mud.
M_2 = position of metacentre corresponding to W_2 L_2

other letters as in previous figures 39, 39a, and 39b.

Taking moments about K.



We have upwards

$$W_2\ (S_I - l) \times a + W_I \times c = W_o \times b.$$

$$\text{but } a = KM_2 \text{ Sin } \theta$$

$$b = KM_3 \text{ Sin } \theta$$

$$c = KM_I \text{ Sin } \theta$$

$$KM_2 \cancel{\text{Sin}}\, \theta \times W_2 (S_1 - 1) + KM_1 \cancel{\text{Sin}}\, \theta \times W_1 = KM_3 \cancel{\text{Sin}}\, \theta \times W_o.$$

M_3 being the resultant of these forces and is the "effective metacentre" Sin θ cancels out.

$$\text{and } Km_3 = \frac{KM_2 \times W_2 (S_1 - 1) + KM_1 \times W_1}{W_o}$$

Total loss of metacentric height

$$= KM_o - KM_3 = M_o M_3$$

but new righting moment

$$= KM_3 \text{ Sin } \theta \times W_o - KG. \text{ Sin } \theta \times W_o$$
$$= \text{Sin } \theta \times W_o (KM_3 - KG) = \text{Sin } \theta \times W_o \times GM_3.$$

and new righting moment $\div W_o = GZ_2$ or new righting lever.

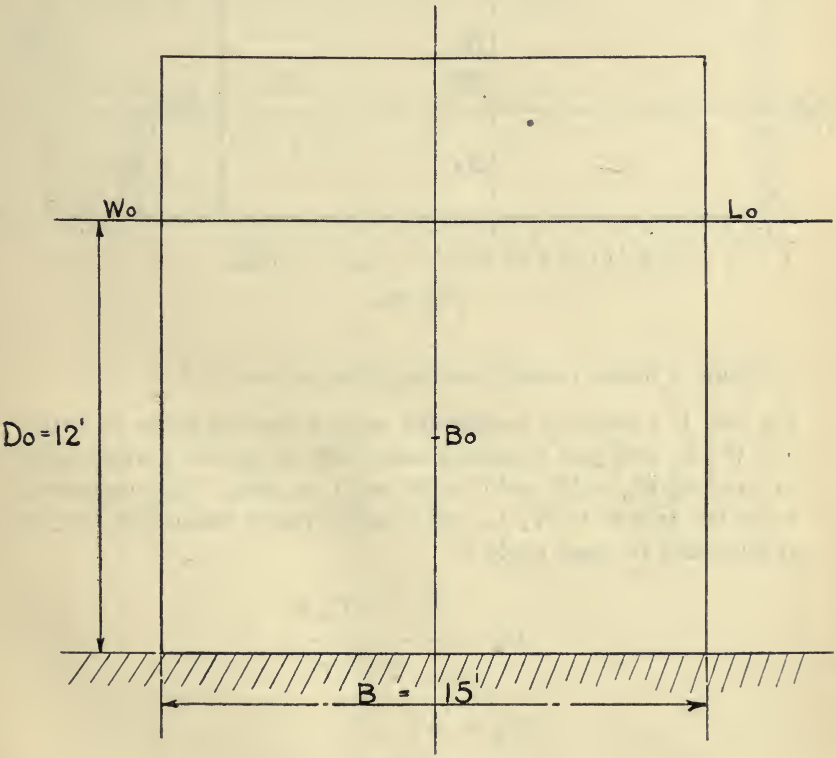


Fig. 39d.

It should be observed that if KG Sin $\theta \times W_o$ is greater than KM_3 Sin $\theta \times W_o$ that GM_3 Sin $\theta \times W_o$ would have a negative sign and an upsetting moment would be present.

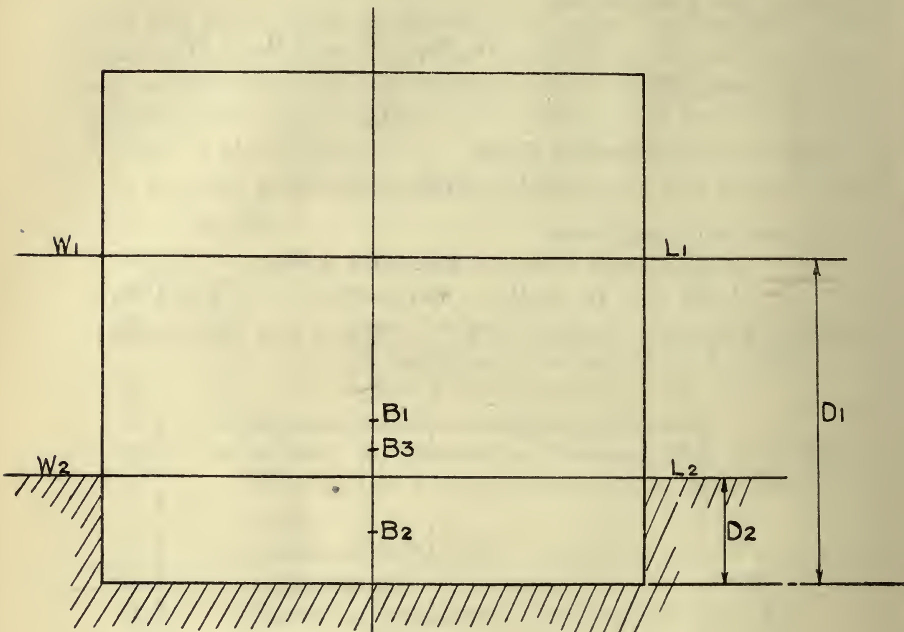


Fig. 39e.

Taking a simple example and applying above :—

Fig. 39d is a vessel of rectangular section floating freely at water-line W_o L_o and just touches a mud bank of specific gravity equal to 2 and say $D_o = 12'$ and $B = 15'$ and $L =$ unity. Now supposing water-line falls 6′ to W_1 L_1 and vessel is partly submerged in mud and inclined to small angle θ

$$D_2 = \frac{12' - (D_2\ 6)}{2 - 1}$$

$$2D_2 = 6$$

$$D_2 = 3' \text{ or } D_1 = 9'$$

$$KM_o = KB_o + B_oM_o = KB_o + \frac{I_o}{V_o} = 6' + \frac{15 \times 15}{12 \times 12}$$

$$= 6' + \tfrac{25}{16} = 7{\cdot}56'$$

$$KM_1 = KB_1 + B_1M_1 = KB_1 + \frac{I_1}{V_1} = 4{\cdot}5 + \frac{15 \times 15}{12 \times 9}$$

$$= 4{\cdot}5 + \tfrac{25}{12} = 6{\cdot}58'$$

$$KM_2 = KB_2 + B_2M_2 = KB_2 + \frac{I_2\ (S_1 - 1)}{V_2\ (S_1 - 1)}$$

$$= 1{\cdot}5 + \frac{15 \times 15}{12 \times 3} = 1{\cdot}5 + \tfrac{25}{4} = 7{\cdot}75'$$

and KG = say 6′0

$$\text{then } KM_3 = \frac{7{\cdot}75 \times 45 + 6{\cdot}58 \times 135}{180} = 6{\cdot}88'$$

$$KM_o - KM_3 = 7{\cdot}56 - 6{\cdot}88 = 0{\cdot}68'$$

CHAPTER VIII.

STATICAL STABILITY AT LARGE ANGLES OF INCLINATION : DEFINITION. ATWOOD'S FORMULA. ORDINARY CURVE OF STABILITY ; CROSS CURVES : REMARKS. INFLUENCING THE SHAPE OF STABILITY CURVES.

Statical Stability at Large Angles. *It is the moment of force (usually expressed in foot-tons) by which a vessel endeavours to right herself, after having been inclined away from the position of equilibrium.* We will now trace the stability of a vessel through the larger angles of inclination where the metacentric method no longer holds good. In Fig. 40 we have a vessel inclined to the angle θ. **W L** was the upright water-line, with **B** its corresponding centre of buoyancy ; $\mathbf{W_1}$ $\mathbf{L_1}$ is the new water-line, which has $\mathbf{B_1}$ as its centre of buoyancy. The displacement is the same up to either water-line. We know that the length of the righting lever **G Z** measured in feet and multiplied by the vessel's displacement in tons gives the amount of foot-tons, moment of statical stability. The displacement for any condition is readily obtained from the displacement scale, and the value of G Z is determined as follows : G Z, we know, is the lever or couple caused by the action of the buoyancy and the weight of the vessel, the buoyancy acting upwards through $\mathbf{B_1}$ and the weight

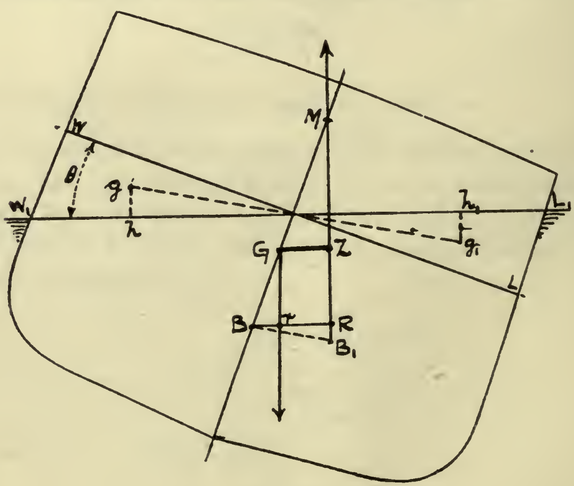


Fig. 40.

acting downwards through **G**. The position of **G** is the same for all inclinations, but **B** alters, therefore for each inclination the value of the shift of **B** must be found. The righting lever being parallel to the water-line and at right angles to the vertical forces, we, therefore, find the shift of **B** in a direction parallel to the water-line corresponding to the inclination. It will be obvious that the shift of **B** is parallel to a line drawn between the centres of gravity of the immersed and emerged wedges; but as we desire it in a direction parallel to the water-line W_1 L_1, we therefore square up on to this water-line the points **g** and $\mathbf{g}_1$, which gives us h and h_1 respectively;

now $\dfrac{v \times h\,h_1}{V}$ = **B R**, which is the shift of the centre of buoyancy parallel to $\mathbf{W}_1$ $\mathbf{L}_1$ or to **G Z**.

v = volume of either wedge.
V = volume of displacement.

Having found **B R**, it is easy to find **G Z**, because on looking into the figure it will be seen that **B R** is larger than **G Z** by the amount of **B r**, therefore

$$B\,R - B\,r = G\,Z.$$

Now $B\,r = B\,G \sin\theta$,

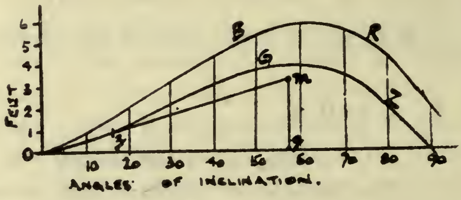
$$\text{therefore } G\,Z = \frac{v \times h\,h_1}{V} - B\,G \sin\theta,$$

this formula being known as **Atwood's Formula.** The minus sign is, of course, only in cases where **G** lies above **B**; but if **G** were below **B**, the formula should be written $\dfrac{v \times h\,h_1}{V} + B\,G \sin\theta$.

$\dfrac{v \times h\,h_1}{V}$ is an equation of form only, and if **B** and **G** coincided, the stability would depend solely on the vessel's form and this equation. The values of $\dfrac{v \times h\,h_1}{V}$ representing B R, having been calculated, they are next corrected for $B\,G \sin\theta$, and then set off and a curve drawn, as shown by G Z in Fig. 41, where the horizontal scale is one of degrees of inclination, the vertical scale

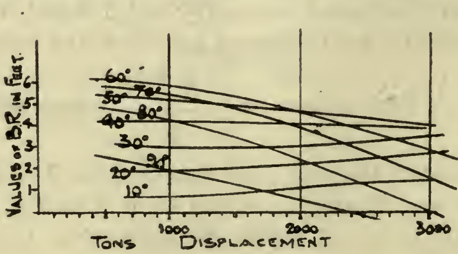
representing the lengths of righting levers in feet. In calculating the stability, it is usual to first of all assume that **B** and **G** coincide, the formula being therefore reduced to $\frac{v \times h h_1}{V}$. This is generally done on account of the position of G not being known at the time of the stability calculations being made. In constructing a curve of stability in this manner, we first of all set off the values of B R on their respective ordinates. In Fig. 41 this curve is shown

Fig. 41.



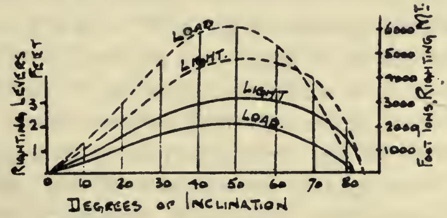
by B R. Next, by setting off values of B G sin θ, corresponding to each angle of inclination used, below the curve B R, **B** being below **G** in this case, we obtain the curve G Z, which is the curve of stability corresponding to the given condition of the vessel. This is known as an **Ordinary Curve of Stability.** To obtain a guide to the shape of the curve at its commencement, the amount of metacentric height, **g m,** is set off at 57⅓ deg. of inclination, and the point **m** joined with **O**, as shown in Fig. 41. At its commencement, the curve will lie upon the line **O m,** as is seen in this Fig. from **O** to z. There are many methods used to find the value of B R, each involving most lengthy calculations, which are unnecessary to produce for our present purpose, a separate calculation being made for a sufficient number of angles of inclination so as to obtain offsets for constructing the curve. An ordinary curve of stability only refers to one particular draught and, therefore, can only be used at this draught.

Fig. 42.



Of course, the amount of a vessel's stability may be required at various draughts, so therefore it is usual to construct **Cross Curves,** from which an ordinary curve for any given draught can be readily obtained. In the case of cross curves, for a base line we have displacements, and the lengths of ordinates being righting levers or moments, or, perhaps, values of B R. Fig. 42 shows a diagram of cross curves. On any one ordinate, say, 1,000 tons displacement is set off the value of B R as found for the various angles used—10, 20, 30, etc., degrees ; this being done on the various ordinates, and then curves passed through their respective offsets so that each curve represents the variation of B R for any one degree of inclination as the displacement alters. Now, suppose that it is required to draw an ordinary curve of righting levers corresponding to a displacement of 1,000 tons, B below G having been ascertained for the particular condition of loading. First of all, in Fig. 41 the base line scale of degrees is drawn, and then the values of B R corresponding to the curves are lifted from the 1,000 ton

Fig. 43.



ordinate of Fig. 53. Transfer these values to their respective ordinates—10, 20, 30 etc., degrees, in Fig. 41, and draw in the curve of B R, below which can then be set off the values of B G sin θ at the various angles, and the curve of righting levers, G Z, thereby obtained. Cross curves are therefore seen to be of great use in enabling an ordinary curve of stability to be quickly constructed for any particular amount and disposition of loading. The righting levers having been ascertained for the given condition, they are next converted into righting moments by multiplying them by the corresponding displacement in tons, which gives the real statical stability of the vessel. In Fig. 43 the full lines represent the curves of righting levers for a vessel in the light and load condition. Converting them into righting moments the curves become as shown dotted. It will be seen that while the lever was largest when light, yet the vessel, in this case, possesses more stability when loaded, the difference in

the relation of the curves being caused by the difference in the displacement.

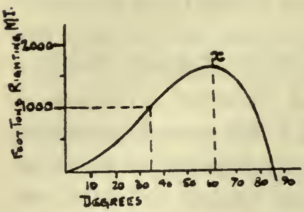


Fig. 44.

Two points often mentioned are: maximum stability and maximum range. The former occurs at the angle where the curve reaches the highest point, as is shown at 62 deg. by **x** in Fig. 44; and the latter is seen to be 85 deg., because the curve here decends below the base line, and the vessel no longer possesses a righting tendency, and, therefore, capsizes. If the vessel is inclined by an external force, such as the wind, it is obvious that when the force is relaxed she will return to the upright if the angle of 85 deg. is not exceeded; but suppose the vessel to be inclined by weights, which are not relaxed, the vessel will then settle down to a steady angle of heel, at which the righting moment of stability equals the upsetting moment caused by the weights. For instance, say 50 tons are moved 20 ft. across the vessel causing an upsetting moment of $50 \times 20 = 1{,}000$ foot-tons, then the angle of rest will be 35 deg., since we here have this upsetting moment balanced by exactly that amount of righting moment. Now, suppose that a larger upsetting moment causes the vessel to incline further than 62 deg. (where we have maximum stability), it is obvious that if the righting moment at x (62 deg.) did not balance the upsetting moment there is no point after that, where it can be obtained, so therefore the vessel will capsize. This instance helps to emphasise the fact that for all practical purposes maximum stability should also be considered as the maximum range.

Influencing the Shape of Stability Curves. A vessel's dimensions control the stability to a large extent, especially in the case of the breadth, which, as previously mentioned, controls the moment of inertia. Breadth has practically no effect upon the height of the centre of gravity, therefore, it only controls the stability due to form. By increasing the breadth the height of M will be increased, and, therefore, the metacentric height will be larger, G remaining unaltered. In Fig. 45 this effect is shown; G Z represents a curve of righting levers, the corresponding metacentric height **g m** being set off at $57\frac{1}{3}$ deg. The breadth being increased results in a larger metacentric height, as shown by **G** $\mathbf{m_1}$. The curve G_1 Z_1 corresponding to the new breadth, lies

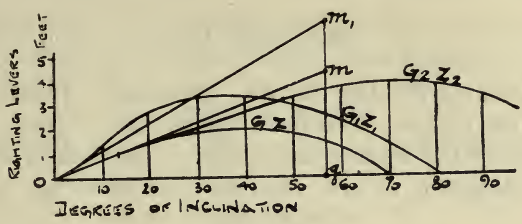


Fig. 45.

upon the line **O m**$_1$ at the origin, the increase in value of the righting levers being in this case more rapid, the curve also showing larger levers and increased range. Altering the depth of the vessel would alter the height of the centre of gravity, and thereby the stability; but suppose that the depth of the vessel is increased, and arrangements made so that the initial stability and draught remain unaltered, the additional depth therefore being added to the freeboard. This additional freeboard is most valuable, the stability being increased so that the maximum lever is of larger amount and takes place at a greater angle of inclination than previously, the range being also lengthened (*see* G_2 Z_2, Fig. 45), the increased freeboard causing the deck edge to become immersed at a later angle. The length of a ship is often said to play no part in the transverse stability, but there is, however, the possibility of it influencing the amount, seeing that in a longer ship the probability is to have a longer 'midship body than in a shorter one, the longer 'midship body having the effect to increase the moment of inertia to such an extent that, even allowing for the increased displacement, the formula $\frac{I}{V}$ may produce a higher position of M for the long vessel. Altering the height of the centre of gravity will obviously influence the stability.

CHAPTER IX.

DYNAMICAL STABILITY: DEFINITION. MOSELEY'S FORMULA. CONSTRUCTION OF CURVE. STABILITY OF SAILING SHIPS; HEELING PRODUCED BY WIND PRESSURE ON SAILS, AND EFFECT OF AREA OF STATICAL STABILITY CURVE. EFFECT OF DROPPING A WEIGHT OVERBOARD.

Definition of Dynamical Stability. "It is the amount of *work done* during the inclining of a vessel to a given angle." We have already seen that the forces which resist the inclination of the ship are vertical forces, the weight of the ship acting downwards and the buoyancy acting upwards with an equal pressure. Throughout an inclination of the ship we find that the centre of gravity is rising or the centre of buoyancy lowering, or both taking place. The total vertical separation of the centres of gravity and buoyancy is the dynamical lever, and this multiplied by the weight of the ship equals the *work done* in foot-tons, or the dynamical stability up to a certain angle. From Fig 40 it will be seen that the vertical separation of G and **B** when inclined to the new water-line $W_1 L_1$ is equal to $B_1 Z - B G$.

Now $B_1 Z = B_1 R + R Z$.

$B_1 R$ being the vertical shift of the centre of buoyancy

$$\text{is} = \frac{v \times (g\,h + g_1\,h_1)}{V}$$

and R Z is equal to r G or B G $\cos\theta$,

$$\text{therefore } B_1 Z = \frac{v \times (g\,h + g_1\,h_1)}{V} + B\,G \cos .\theta$$

Dynamical lever or vertical separation:

$$B_1 Z - B\,G = \left\{ \frac{v \times (g\,h + g_1\,h_1)}{V} + B\,G \cos\theta \right\} - B\,G.$$

We know that the difference between 1 and $\cos\theta$ is versin θ ($1 - \cos\theta$ = versin). It will therefore be seen that the difference

between B G and B G cos θ will be B G versin θ, and this being the amount by which the *minus* quantity is in excess, we now write :

$$\text{Dynamical lever} = \frac{v \times (g\,h + g_1\,h_1)}{V} - \text{B G versin } \theta,$$

which is the vertical separation of **G** and **B.**

$$\text{Dynamical moment} = W\left[\frac{v \times (g\,h + g_1\,h_1)}{V} - \text{B G versin } \theta\right]$$

which is the *work done.* This is known as Moseley's Formula.

The dynamical Stability up to any angle, or between any two angles, is equal to the area of the curve of statical stability for the corresponding distance of inclination, therefore, by calculating the area of a statical curve up to a few inclinations, and setting the results up on their respective ordinates, we can obtain the points through which to draw the curve of dynamical stability, which is shown by D S in Fig. 46.

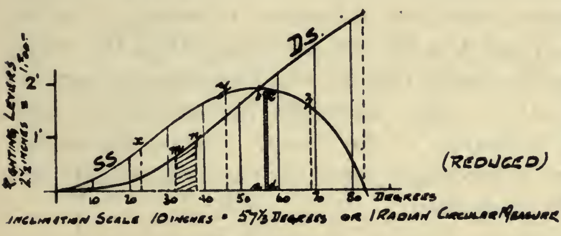


Fig. 46.

It will be noticed that the dyanmical curve reaches its maximum when the statical curve, S S, reaches zero. By subtracting one ordinate from another, such as **n** — **m,** we can obtain the amount of work that is exerted in inclining the vessel from **m** to **n**, or otherwise the amount of dynamical stability. Between any infinitesimally small strip, such as **a b c d**, Fig. 46, the dynamical moment will equal the statical moment **a b** multiplied by the circular measure of the small angle contained in the strip, this also giving the area of the strip. From this it will be seen that it is necessary to make the scale of inclination also in circular measure, or at least to regard it as such in the calculation. Suppose we had given the curve S S, Fig. 46, of righting levers, the scales being as shown. First of all, take an ordinate **x**, and by means of Simpson's rule, or, preferably, the planimeter, find the area in square inches, of the curve up to that point. We have next to take into consideration the scales that

the curve is drawn to, this being done as follows: Find what 1 in. is equal to in each scale, and by multiplying these together we then have a multiplier with which to multiply the area already found, and we then have the dynamical lever in feet, which, when multiplied by the vessel's weight, gives the dynamical moment in foot-tons. Suppose that the area up to **x** was found to be 10 square in. actual area of the curve, and the weight of the vessel 100 tons.

Vertical Scale, $2\frac{1}{2}$ ins. = 1 ft. G Z
$\therefore$ $\frac{1}{4}$ in. = ·1 ft. G Z and 1 in. = ·4 ft. G Z.

Inclination Scale, 10 in. = $57\frac{1}{3}$ deg.
$\therefore$ 10 in. = 1 radian and 1 in. = ·1 radian,

therefore 1 sq. in. = ·04 foot-radians.

10 sq. in. Area × ·04
= ·4 ft. dynamical lever.
·4 ft. × 100 tons = 40 foot-tons dynamical moment.

Having in this manner found the dynamical moments up to the ordinates such as **x**, **y** and **z** in Fig. 46, and then setting the values off on them, we have the points through which to draw the curve of dynamical stability D S, which represents the amount of work exerted during the inclination of the vessel. This is the method usually employed in constructing a curve of dynamical stability and is known as graphic integration.

Stability of Sailing Ships. In Fig. 47 the section is shown of a vessel with some sail set. The centre of gravity of the sail area is at **E**, which is known as the centre of effort, through which the total pressure **P** of the wind may be said to be acting. Upon the lee-side we have a pressure of water, **p** acting in the opposite direction to the wind pressure. This water pressure acts through the centre of lateral resistance, which may be taken to be the centre of gravity of the under-water middle-line vertical plane. When **P** and **p** are equal the vessel makes no leeway. In Fig. 47 the distance vertically between the centre of effort and centre of lateral resistance = x feet. The pressure of wind upon the vessel's sails = P tons. Therefore the inclining moment = (P × x) foot-tons. Now, this moment inclines the vessel over to the angle θ, at which we have a righting lever **G Z**, therefore

P × x (inclining moment) = W × G Z (righting moment), W being the total weight of vessel.

P, the pressure on the vessel's sails, is effected by sail area and wind pressure ; x is effected by the distribution of the sails. The area is that taken from the projection of the sails on to the centre line of the vessel. It is obvious that as the vessel inclines this projected area will become less. Notice in the adjoining sketch in Fig. 47 that when **m** is inclined to angle θ the vertical distance **A** becomes $\mathbf{A_1}$. Now, suppose that **A** represented the sail area in square feet when upright, then $\mathbf{A_1}$ will equal the projected area when inclined to θ, being equal to A $\times$ cos θ. Let **x** be the distance from the centre of effort to centre of lateral resistance when upright, then $x \times \cos\theta$ will be the vertical distance when inclined to θ.

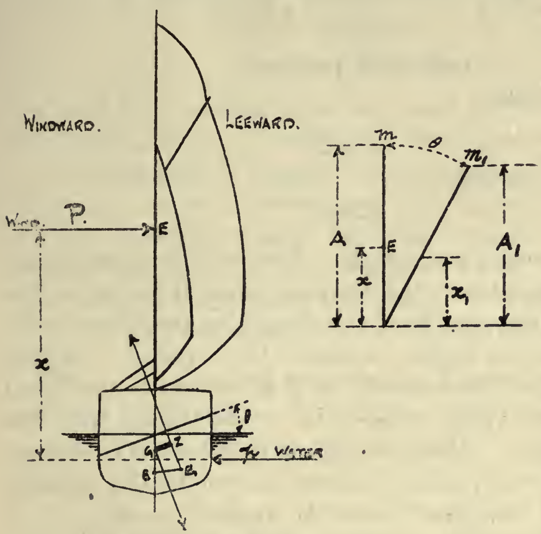


Fig. 47.

Heeling mement when upright =
Tons per sq. ft. wind pressure $\times$ A $\times$ x = M.
Heeling moment when inclined =

Tons per sq. ft. wind pressure $\times$ A cos θ $\times$ x cos θ = M cos² θ. The effect of a sudden gust of wind upon the sails of a ship will, as a rule, heel her over to an extreme angle of heel of about twice the steady angle at which the same constant pressure of wind would keep her. The following method is one by means of which we can determine the angle of extreme roll and the ultimate angle of steady heel of a ship when struck by a squall of known force. In Fig. 48 we have a curve of statical stability represented by S.S. Upon the same diagram, and to the same scales, proceed to construct the curve of varying wind pressures as follows :—First of all find the area of the sails and the position of the centre of effort relative to the centre of lateral resistance. Next assume the wind pressure of, say, 2 lbs. per sq. ft., which is a very common figure to take for

these calculations, with all canvas set, at which time the vessel should possess a reserve and the angle of lurch should not exceed the angle of maximum G Z.

Multiplying the area in square feet by 2 lbs. and dividing by 2,240, we obtain the total number of tons pressure on the vessel's sails when upright

$$\frac{A \times 2}{2{,}240} \text{ tons wind pressure.}$$

this, multiplied by x in feet $= \dfrac{A \times 2}{2{,}240}$ tons $\times$ x feet $=$

foot-tons heeling moment when upright. Now, we have already seen that the heeling moment or wind pressure moment varies as the $\cos^2\theta$ during inclination, therefore, by multiplying the above result by $\cos^2\theta$ for the various angles, we obtain the curve of "varying wind pressure moments," as shown by **W P M**. Suppose the squall suddenly to strike the vessel, causing her to heel away from the upright towards leeward. When the angle of inclination **a** is reached, it will be noticed that the righting moment and the heeling moment are equal; this must, therefore, be at the angle of steady heel at which the vessel would sail if the wind pressure were steadily applied. The wind pressure moment up to **a** is represented by **c d e a**, part of which is counter-balanced by the area **c e a** of the righting moment curve **S S**. We therefore have left the portion of the wind pressure moment **c d e**, which yet remains to be counter-balanced by righting moment; therefore, on this account, the vessel continues the inclination until a point is reached where the excess **c d e** is counter-balanced by an area of the S S curve lying above the W P M curve, as is shown by the area **e f g**. In Fig. 48 this point is the angle of inclination **b**, which is, therefore, the angle of extreme roll, up to which the wind force has absorbed an equal amount of the vessel's righting force, being equal to the *work done.*, After reaching the angle **b** the vessel returns to the angle **a**, at which she settles down until the force is relaxed. Now, suppose that the curve of statical stability had taken the shape as shown by S_1 S_1; it will be noticed that the portion of the righting moment curve area above the W P M curve would not be sufficient to counter-balance the excess **c d e** of wind pressure moment, therefore the vessel would capsize because

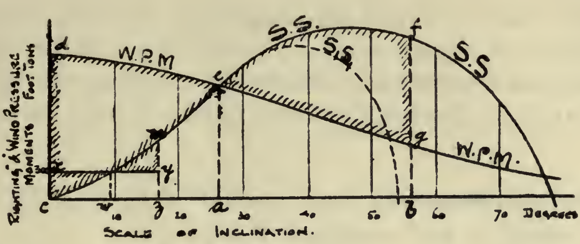


Fig. 48

the total force exerted upon her could not be counter-balanced by her righting force. A large area of the curve of statical stability is, therefore, seen to be most valuable in the case of sailing vessels. Of course it must be remembered that in the above cases no account is taken of fluid resistance upon the immersed portion of the hull or air resistance of the portions above water, the deductions by the above method being purely theoretical. The effect of the resistances, however, may be regarded as a margin of safety.

Effect of Dropping a Weight Overboard. If a weight be suddenly dropped from off one side of a vessel—such as, for instance, a weight falling owing to the breaking of a rope suspended from a derrick—we have an effect similar to the above suddenly applied force. Say a weight of 10 tons, suspended at 30 ft. from the centre line, were to fall in such a manner, a force of $10 \times 30 = 300$ foot-tons would be suddenly *withdrawn* from one side of the vessel. To find the angles of extreme roll and steady heel, the line **x y** is drawn in Fig. 48 at the corresponding moment of 300 foot-tons. At the angle **z** the inclining moment represented by the area **c x y z** is counter balanced by the area **c m z** of the righting moment curve, **z** therefore being the angle of theoretical extreme roll. At **w** it will be seen that the inclining moment and righting moment are equal, this therefore being the angle of steady heel.

CHAPTER X.

LONGITUDINAL STABILITY: METACENTRE AND CALCULATION. TRIM AND MOMENT TO CHANGE TRIM ONE INCH. ESTIMATIMG THE TRIM.

Longitudinal Metacentre and method of calculation. Fig. 49 shows a vessel in two conditions of trim.

B = Centre of Buoyancy } Corresponding to the upright water-
G = Centre of Gravity } line W.L.
B₁ = Centre of Buoyancy } Corresponding to the inclined water-
G₁ = Centre of Gravity } W_1 L_1.
CF = The centre of Flotation, being the point of intersection of the two water-lines.
g and **g_1** = Centres of Gravity of emerged and immersed wedges respectively.
M_l = Longitudinal Metacentre.
w = A weight on board which is moved aft, causing the vessel to change trim sternwards.
d = distance **w** is moved.

Let V = the volume of displacement, and v the volume of either wedge, then—

$$\frac{v \times g\, g_1}{V} = B\, B_1, \text{ and}$$

$B\, B_1 = B\, M_l \times$ circular measure of the angle of inclination.
(Let θ be the angle of inclination, being equal to either **a**, **b** or **c**.)

$$\text{Therefore } B\, M_l \times \text{circ. } m\, \theta = \frac{v \times g\, g_1}{V}$$

$$\text{and } B\, M_l = \frac{v \times g\, g_1}{V \times \text{circ. } m\, \theta}$$

At A take an infinitesimally small transverse slice of the wedge, assuming **d x** to be its breadth in a fore and aft direction, and the length across to be **y**. The depth of the slice will be $x \times$ circ. $m\, \theta$ and its volume will be $y \times dx \times x \times$ circ. $m\, \theta$. The moment of this small volume about C F will be

$$(y \times dx \times x \times \text{circ. } m\, \theta) \times x = x^2 \times y \times dx \times \text{circ. } m\, \theta.$$

If all such moments throughout the length of the wedges were summed up, we would obtain the total moment of transference—viz., $v \times g\,g_1$, which is therefore equal to

$$\int x^2 \times y \times dx \times \text{circ. } m\,\theta.$$

This can now be substituted for $v \times g\,g_1$.

$$BM_l = \frac{v \times gg^1}{V \times \text{circ. } m\theta} = \int \frac{x^2 \times y \times dx \times \cancel{\text{circ. } m\theta}}{V \times \cancel{\text{circ. } m\theta}}$$

Circ. m θ cancelling out, we have $BM_l = \int \dfrac{x^2 \times y \times dx}{V} = \dfrac{I}{V}$

CALCULATION FOR LONGITUDINAL METACENTRE.

No. of Ordinates.	Lengths of Half-Ordinates.	S. M	Functions.	Levers.	Products for Moments.	Levers.	Products for Moment of Inertia.
0 aft		1		4		4	
1		4		3	Moments aft	3	
2		2		2		2	
3		4		1		1	
4		2		0	sum = 297·28 aft	0	
5		4		1		1	
6		2		2	Moments forward	2	
7		4		3		3	
8		1		4		4	

Sum of Functions = 367·18 <——
× ⅓ longitl. interval = 12·5

4589·75

For both sides × 2

Waterplane area = 9179·50 s. ft

Sum = 282·56 ford
,, = 297·28 aft

Excess = 14·72 aft
× Longitl. interval 37·5

——> 367·18)552·00

Centre of flotation aft of amidships = 1·5 ft.

Sum = 1272·92
× ⅓ longitl. interval 12·5

15911·5
× Longitl. interval squared $37·5^2$

22375551
For both sides × 2

44751102
——> Correction — 20654

÷ By volume of displacement)44730448(MI)

Longitudinal Metacentre above CB = 487·8 ft.
Centre of buoyancy above base = 8·52 ft.

Longitudinal Metacentre above base = 496·32 ft.

Correction for moment of inertia about an axis passing through the centre of flotation = area of waterplane × distance of C F from 'midships squared—
$9179·5 \times 1·5^2 = 20654$ to be deducted in all cases). <—

It will be seen that the numerator is the algebraic way of expressing moment of inertia, because we have the small areas $y \times d\,x$ multiplied by the distance squared that their centres are from the axis.

In the calculation the area and centre of flotation of the water-plane is first found in the usual way, and then the products for moments are again multiplied by the levers, the products for moments of inertia being thereby obtained since the functions of the half-ordinates are now multiplied by their levers squared. The sum of the products for M I is multiplied by $\frac{1}{3}$ of the common longitudinal interval, so as to complete Simpson's Rule, and then by the interval squared, the latter being necessary to convert the leverages into actual distances which are squared. Multiplying by two for both sides, we have then the longitudinal M I about the 'midship ordinate, but as the centre of flotation is the correct axis, a correction is next made as shown.

Trim. By "trim" is meant the amount of change of draught aft plus the change of draught forward. In Fig. 49, $W_1\ W + L\ L_1$ = Trim. The tipping centre is taken as being the centre of flotation. Change of trim is caused in the following ways: first, by moving a weight, already on board, in a fore and aft direction; second, by

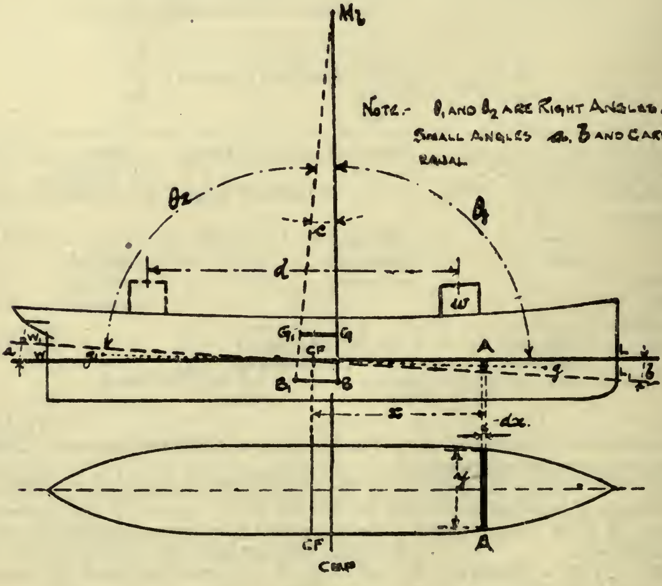


Fig. 49.

placing a weight on board; third, by taking a weight off a vessel. In the second and third we have sinkage and rise respectively, as well as change of trim, to take into account. In the first, we have a weight **w** in tons moved a distance **d** (*see* Fig. 49). **w** × **d** = the trimming moment in foot-tons. Now, if we know the amount of moment necessary to alter the trim 1 in (*i.e.*, ½ in. at each end), we can find the total amount of change of trim caused by the shifting of a weight already on board the vessel thus: Trimming moment ($w \times d$) ÷ moment to change trim 1 in. (M C T 1 in.).

Moment to Change Trim 1 in.

In Fig. 49, $\dfrac{w \times d}{W} = G\,G_1$ (W being the total weight of the vessel),

and $G\,G_1 = G\,M_l \tan\theta$

or $\dfrac{w \times d}{W} = G\,M_l \tan\theta$,

$$\therefore \frac{w \times d}{W \times G\,M_l} = \tan\theta.$$

The total trim $= W\,W_1 + L\,L_1 = T$.

L (length of water-line) $\times \tan\theta = T$, or $\dfrac{T}{L} = \tan\theta$,

$$\therefore \frac{w \times d}{W \times G\,M_l} = \frac{T}{L}$$

If the change of trim is 1 in., then

$$\frac{w \times d}{W \times G\,M_l} = \frac{1}{L \times 12}$$ (L being in feet, must necessarily be multiplied by 12).

$\therefore \dfrac{W \times G\,M_l}{L \times 12} = w \times d$, the trimming moment where the trim is 1 in., so we have M C T 1 in., $= \dfrac{W \times G\,M_l}{L \times 12}$ W being the

total displacement, G M_l the height of the longitudinal metacentre above the centre of gravity, and L the length of the water-line.

Dividing the trimming moment by M C T 1 in. we have the amount of trim that is to be divided at each end according to the position of the centre of flotation, which is the tipping centre. If the tipping centre is at 'midships, the amount of trim must be divided equally over each end of the vessel. When the alteration is small, the tipping centre may be assumed to be at 'midships ; but when the change is large, it must be proportionately distributed according to the position of C F. For instance, suppose a vessel 100 ft. long with C F at 25 ft. aft of 'midships and the total change of trim to be 12 in. in a sternward direction, the relative changes would be

$\frac{25}{100}$ of 12 in. $=$ 3 in. trim aft (increase of draught),

$\frac{75}{100}$ of 12 in. $=$ 9 in. trim forward (decrease of draught).

Adding a Weight. First, take the case of a weight of moderate amount. If it were desired to place it on board in such a position so that the trim would remain unaltered, it would have to be placed so that the downward force of the added weight would be acting in the same vertical line as the upward force of the added buoyancy contained in a parallel layer—that is, the centre of gravity of the added weight would be placed over the centre of buoyancy of the added parallel layer of buoyancy. In this case, the weight being of a moderate amount, the layer would be a thin one and will have its centre at a position just about coinciding with the centre of flotation of the former water-line, so, therefore, we may say that, for moderate amounts of weight placed vertically over or under the centre of flotation, we have only sinkage and no change of trim. The sinkage is obtained by dividing the amount of added weight by the tons per inch corresponding to the water-line.

We therefore see that the following assumptions are made : First, the tons per inch immersion alters only very slightly as the draught increases ; and second, the centre of buoyancy of the added layer is in the same vertical line as the centre of flotation of the former water-line. Now, having found the position, as above, in which to place the weight so as not to alter the trim, we can then proceed to find the trimming moment by multiplying the added weight by the distance of its centre from the centre of flotation, and this moment divided by the M C T 1 in., will give the trim. Now, suppose the

weight to be of a large amount. We will have to discard the assumptions made in the previous case, and take the following into account :

1. To find the new parallel draught from the displacement scale by adding the amount of the weight to the original displacement, and then to read off the new mean draught corresponding to the new total displacement.

2. The correct position of centre of buoyancy of added layer.
3. The correct position of centre of flotation of new waterplane.
4. The vertical position of **G** will be altered most probably.
5. The increase of draught will alter the position of $\mathbf{M}_l$.

The 4th and 5th will affect the M C T 1 in. The position of the centre of buoyancy of the added layer of displacement is found as follows : From the curve of centres of flotation read off the position of C F for a waterplane at half-depth of the layer between the old and the new parallel draught, and this position may be used as the centre of buoyancy of the layer. The C F of the new parallel waterplane is also found from this curve. Now find the position of **G** after the weight has been added, and the new position of $\mathbf{M}_l$ can be taken from the curve of longitudinal metacentres, the new metacentric height being thereby obtained which enables us to find the new M C T 1 in. The trimming moment is equal to the added weight multiplied by the distance of its centre from the C B of the added

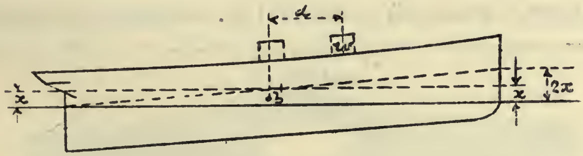


Fig. 50.

layer, and this moment, divided by the new M C T 1 in., gives the amount of trim to be distributed according to the position of the centre of flotation (tipping centre) of the new waterplane.

Deducting a Weight. In this case the method above described is simply reversed. The depth of deducting layer of displacement and its C B and also the C F of the new waterplane are found in the same manner. Multiplying the weight by the distance that its centre was from the C B of the deducted layer we then have the trimming moment, which, when divided by the M C T 1 in., as

corrected for the new condition, will then give the total amount of trim to be distributed according to the position of C F of the new waterplane.

To obtain the position in which to place a weight so that the draught aft will not be altered by means of its addition, deduction or gradual consumption. Such a position is often required in connection with a vessel's bunkers, so that the propeller immersion will remain constant during fuel consumption. First find the sinkage if the weight is placed vertically in line with **b**, the centre of buoyancy of the added layer of displacement (*see* Fig. 50). This sinkage is **x**, being the increase of draught at each end, since the layer is parallel. The draught aft in this condition will be exceeding the requirements by the amount **x**, therefore the vessel's trim requires to be now amended so that the draught aft will be reduced to its former amount. The vessel may be assumed to tip about 'midships, and the required alteration being a reduction of **x** draught aft and a further increase of **x** forward, the total change will be 2**x** in a forward direction. The total change of trim in inches, when multiplied by the M C T 1 in. corresponding to the condition, will give the amount of trimming moment required to produce this change. This trimming moment being divided by the known weight **w** will then give the distance **d** forward of the C B of the layer, in which to place the weight so as to obtain the required trim :

$$\therefore \frac{(x \text{ sinkage in ins.} \times 2) \times \text{M C T 1 in.} = \text{trimming moment}}{\text{weight } w} = d$$

CHAPTER XI.

RESISTANCE. WATER RESISTANCE AND STREAM LINE THEORY. FRICTIONAL RESISTANCE, EDDY-MAKING RESISTANCE, WAVE-MAKING RESISTANCE AND AIR RESISTANCE. MODEL EXPERIMENTS. HORSE POWER. EFFECTIVE H.P. INDICATED H.P. PROPULSIVE CO-EFFICIENTS. LOSSES. COMPOSITION OF I.H.P. NOMINAL H.P.

Tow Rope Resistance. The total resistance of a ship is composed of Skin Friction (R_s), Eddy-making (R_e), Wave-making (R_w), and Air Resistance (R_a). Total Tow Rope Resistance $= R_s + R_e + R_w + R_a = R$. If a ship were towed in such a manner so that the presence of the tug would not affect the resistance of the ship, then the ship would impart to the rope a strain equal to R. This resistance is augmented in the case of a vessel that is propelled by her own means, such as screw propeller, etc. In dealing with resistance, it is usual to disregard this propeller augment, and only work with the tow rope resistance or "resistance of form," allowance being afterwards made for this augment when estimating the horse-power.

Resistance of the Water. Water is not a perfect fluid ; when in motion its particles rub against each other, causing an amount of friction which is known as viscosity. When rubbing against a body such as a ship, an amount of frictional resistance is set up. If a ship-shaped body, as shown in Fig. 51, is submerged and drawn

Fig. 51.

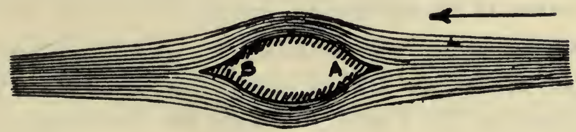
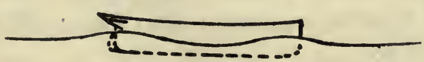


Fig. 52.



through the water, or, say, fixed in a position and the water allowed to flow past it (which has exactly the same effect), the particles of water are diverted in the directions as shown by the lines outside the figure, these lines being termed stream lines. The direction of the flow of water is shown by the arrow. When nearing the body, the stream lines widen out, reaching a maximum width at the end

A. This widening causes a decrease in the speed of the flow at this point, and, consequently, an increase of pressure. At 'midships the distances between the stream lines become narrower, which now causes an increased speed, giving a reduced pressure. Upon reaching the aft end **B** a similar effect is obtained as at the fore end **A.** The effect of this change of pressure can be noticed in a vessel moving upon the surface, where the increase of pressure is seen to cause the water to increase in height at the ends, while the reduced pressure at 'midships causes a reduction in height (*see* Fig. 52), the two waves being known as the "statical crests." The stream lines of water flowing past the hull of a vessel obtain complicated motions, the particles becoming confused with each other, and, rubbing against the hull, result in the frictional resistance, the amount of which is dependent upon the speed, the roughness of the immersed surface of the vessel, its area, and the length. The late Dr. Froude performed interesting and valuable experiments in connection with skin frictional resistance upon plane surfaces of varying lengths and nature of surface, by towing them in the experimental tank at Torquay, obtaining results at various speeds. In estimating this **Frictional Resistance,** the length, area and nature of surface, speed at which it is travelling, and the density of the water are taken into account, the calculation being made by using data obtained from such experiments, in connection with the following formula:

$$R_s = f\ S\ V^n,$$

R*s* being the skin fractional resistance of the vessel in pounds.

f, a co-efficient varying according to the length of the vessel and also the nature of the surface.

S is the area of the wetted surface in square feet, being found as described in Chapter III., or by the formula $15{\cdot}5\ \sqrt{W \times L}$, which is given by Mr. Taylor. W being the displacement in tons and L the length of the vessel. Fairly good results are quickly obtained by this formula, especially when the co-efficient is obtained from a similar ship.

V is the speed in knots per hour.

n, an index accounting for the variation of the resistance with the speed.

For steel plated vessels in salt water, with the immersed surface painted and in clean condition, the following figures for *f* and *n* may be used.

Length of ship	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.
f	·0097	·00944	·00923	·0091	·00904
n	1·83 in all cases.				

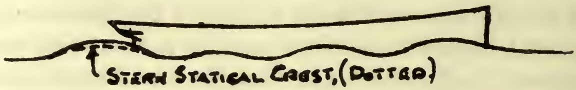
These figures will also take into account the eddy-making resistance for vessels of ordinary and good form. For a vessel 350 ft. long, with a wetted surface area of 24,500 sq. ft., the skin frictional and eddy resistance when moving in salt water at the speed of 11 knots per hour will be $\cdot 00916 \times 24500 \times 11^{1\cdot 83} = 18{,}064$ lbs. At first sight it may appear that the square of the speed may be used instead of the 1·83 power, the latter seeming to be near to the square ; this, however, is far from being admissable, and 1·83 must be strictly adhered to. At low speeds the frictional resistance forms a large proportion, being about 80 to 90 per cent. of the total at speeds of 6 to 8 knots in vessels with clean bottoms. At about 20 knots, when wave-making is playing an important part, the frictional resistance is about 45 per cent. to 60 per cent. The frictional resistance being of such a large proprotion at the slower speeds, it is obvious that economy is obtained in full-lined vessels of slow speed, in which a larger proportion of displacement to wetted surface is obtained, than in the case of fine-lined vessels.

Eddy-making Resistance. Blunt endings of the immersed body of a vessel, such as full sterns and broad stern-posts, cause eddies which draw upon the power of the vessel in the shape of setting up extra resistance to be overcome. Such endings should, therefore, be avoided as far as possible. In modern, well-formed vessels eddy making is of small amount, and is usually taken as being 5 per cent. of the skin frictional resistance. However, in the figures above for f and n allowance is made for the extra 5 per cent. and when these figures are used, no separate calculation is required for the eddy-making.

Wave-making Resistance. In addition to the statical crests formed at the bow and stern, as already mentioned, other waves of reduced height will also be generally noticed ; a series of waves will be found abaft the bow wave and another series abaft the stern wave. The two series being of distinctly separate creation, an interesting relation is, therefore, obtained when the bow series joins the stern series. Should it happen that a crest of the bow series coincides with the stern statical crest, the wave of the former is piled up on top of the latter, as shown in Fig. 53, causing a large

stern crest with a following train of large waves, the wave-making resistance being greatly augmented thereby. If a hollow of the bow wave series had coincided with the stern statical crest there would have been a counteracting effect, and consequently fairly flat water astern resulting in a reduction of wave-making resistance, The distance in feet, from crest to crest of the bow wave series in deep water, is found to be about ·56 V^2 (V being the speed in knots per hour). By use of this formula a vessel can, therefore, be designed

Fig. 53.



so that, at her working speed, the hollow of a bow wave will coincide with the stern statical crest. The distance between the statical crests in fine ships is about equal to the water-line length ; in full ships the distance being about 1·1 times the water-line length. In a fine-line vessel of 20 knots, the first echo of the bowwave will be $·56 \times 20^2 = 224$ ft., which would, of course, be a very bad length for a vessel of this speed, while 336 ft. would be very good, since we would, in this case, obtain a " bow series hollow " coinciding with the stern statical crest. Propellers, or abnormal form of lines, may modify these natural waves, or even create additions. If a vessel is run at varying speeds, and a curve of corresponding horse-powers constructed, it is generally found to be of an uneven character, as shown in Fig. 54, this being caused by the above-mentioned maximum and minnimum wave effects, the humps shown by **h** being at speeds where the stern statical crest is augmented by a wave of the bow series. The following formula, given by Mr. Taylor, may be used :

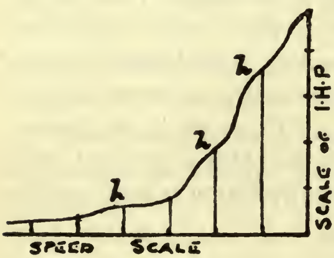


Fig. 54.

Wave-making Resistance in lbs.

$$(R_w) = 12·5 \times C \times \frac{D}{L^2} \times V^4$$

V = speed in knots per hour, D — displacement in tons, L = length in feet, and C = block co-efficient.

Air Resistance. Air resistance is only of importance at high speeds, or when a vessel is steaming against a head wind. At low speeds it is generally neglected because of its small amount, but at high speeds it becomes worthy of notice, and is usually allowed for to the following extent:

$$\text{Air Resistance in lbs. } (R_a) = \cdot 005\ A\ V^2$$

A being the area of the hull and erections exposed ahead, and V the speed, in knots per hour, of the air past the vessel.

Model Experiments. To calculate resistance in this way a model of the proposed vessel is towed in a tank, the speeds and corresponding resistances being recorded and then proportionately increased to suit the vessel's dimensions. In converting the results of a model experiment "Froude's Law of Comparison" is used. According to this law, the *corresponding speeds* of two ships, or a ship and a model, vary as the square root of the length ratio

$$\sqrt{\frac{\text{ship length}}{\text{model length}}} = \sqrt{l} = \text{variation}$$

of speeds, therefore the speed of the model multiplied by $\sqrt{l}$ will give the *corresponding speed* of the ship; also, at *corresponding speeds*, the residuary resistance (*i.e.*, resistances other than the Skin friction) will vary as the length ratio cubed: $\left(\frac{\text{ship length}}{\text{model length}}\right)^3 = l^3$.

It should be remembered that skin frictional resistance does not follow the "Law of Comparison." In Fig. 55, **A** represents a curve of total resistance in lbs., as constructed from the recorded results of experiments made upon a model towed at various speeds. By means of the formula $f\ S\ V_n$, the frictional resistance of the model at various speeds is calculated, and deducting the amounts from the curve **A**, the curve **B**, which is residuary resistance, is obtained. The "Law of Comparison" applies to this curve, and by this means it is converted to suit the ship. By

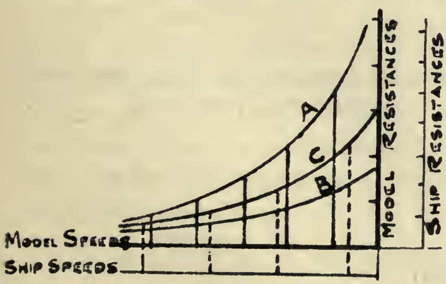


Fig. 55.

simply altering the scales of speed and resistance the curve **B** can be used for the ship, therefore by multiplying the model speeds by $\sqrt{l}$ a new scale of *corresponding speeds* for the ship is obtained, and by multiplying the model resistance scale by l^3 we obtain a scale of ship resistance, a further multiplication of 1·026 being necessary if the experiment was made in fresh water and the ship's resistance is required in salt water. These two new scales being annexed to the diagram, we can readily obtain the residuary resistance of the ship from the curve **B**, but the total resistance being required, the skin frictional resistance is next calculated for the ship, and the amounts added to the curve **B** gives **C**, which is the curve of total resistance of the ship. **Co-efficient f and index n for paraffin wax models in fresh water:**

Length.		f.		n.
10 ft.	...	·00937	...	1·94
11 ft.	...	·00920	...	,,
12 ft.	...	·00908	...	,,

Horse Power. The unit of *force* taken as being 1 lb., and the unit of *speed* as 1 ft. per minute, gives the unit of power as follows:

Force × Speed = Power.

1 lb. × 1 ft. per minute = 1 ft. lb. per minute (the unit of power).

33,000 of such units are regarded as 1 *horse power*. In the case of a ship the resistance is the force; therefore, if R is the resistance in lbs. and v the speed in feet per minute, then the horse power required to overcome this resistance will be:

$$\frac{\mathrm{R} \times v}{33000}$$

or, if we deal with speed in knots per hour, as this is the usual way of expressing it, and which is represented by V, we have:

$$\text{Horse power} = \frac{\mathrm{R} \times \mathrm{V} \times \frac{6080}{60}}{33000} = \frac{\mathrm{R} \times \mathrm{V} \times 101{\cdot}33}{33000}$$

$$= \mathrm{R} \times \mathrm{V} \times {\cdot}0030707$$

The latter is a much handier expression, and is obtained by converting the speed of knots per hour into feet per minute by multi-

plying by the number of feet in 1 minute for the rate of 1 knot per hour, which is 101·33. This being divided by 33000 gives ·0030707, which now gives a much simpler and reduced formula.

Effective Horse Power. This is the amount of power that is consumed in the actual propulsion of the vessel, being the amount necessary to overcome the total of the various resistances which have been previously dealt with—*i.e.*, the tow rope resistance. It may be termed the tow rope horse power, because it is equal in amount to that which would be transmitted through a tow rope when towing the vessel at the given speed, the effect of the presence of the tug being, of course, eliminated. On account of power being consumed by means of friction of the machinery, working of auxiliaries off the main engines, inefficiency of propeller, etc., an amount of power much larger than the E H P must be generated in the vessel's engines, since the E H P is simply the amount which is usefully employed in overcoming the external resistances to the vessel's motion.

Indicated Horse Power. This is the amount of power that is actually generated in the vessel's engines, being measured from the engines by the instrument known as the indicator, from which the name is derived. By use of the Indicator, diagrams are obtained which graphically show the variation of the steam pressure in the cylinders, the mean pressure being calculated from such diagrams. If P = the so found mean pressure in lbs. per square inch and A = the area of piston in square inches, we have P A = the total mean pressure (*i.e.*, the *force*) upon the piston in lbs. This pressure multiplied by the speed will equal the *work done*, which when divided by 33,000 gives the I H P. The speed is equal to the distance run by the piston in 1 minute, which is as follows : If L = the length of the stroke of the piston in feet, then during one revolution the distance will be 2 L, and if N is the number of revolutions per minute, the distance run in 1 minute will be 2 L N. We therefore now have a *force* of P A pressure in lbs., and a *speed* in feet per minute of 2 L N.

$$P\ A \times 2\ L\ N = \text{ft. lbs. per minute (units of power),}$$

$$\text{or, } \frac{P\ A \times 2\ L\ N}{33000} = \text{indicated horse power.}$$

A simple way of remembering this is by constructing the word "Plan" from the above letters, viz.:

$$\frac{2\,P\,L\,A\,N}{33000} = I\,H\,P.$$

Indicated horse power will be seen to vary as the steam pressure P, the cylinder capacity L A, and the number of revolutions N.

Propulsive Co-efficients. The ratio of the E H P to the I H P is known as the "Propulsive Co-efficient": for instance, in the case of a vessel where 1,000 I H P is necessarily provided to drive her at a speed at which the E H P is 600, the propulsive co-efficient is

$$\frac{E\,H\,P}{I\,H\,P} = \frac{600}{1000} = \cdot 6.$$

It is sometimes expressed as a percentage, which, in this case, would be 60. This co-efficient varies at the different speeds of a vessel, owing to the fact of some of the losses being nearly constant in amount, therefore absorbing a larger proportion of the I H P at the slower speeds, and also owing to the propellers being designed to give the maximum efficiency at the designed top speed of the ship. The higher the propulsive co-efficient the greater is the efficiency, therefore the co-efficient should reach its maximum at the designed top speed, a curve of such co-efficient giving a shape as shown in Fig. 56. The following are typical values:

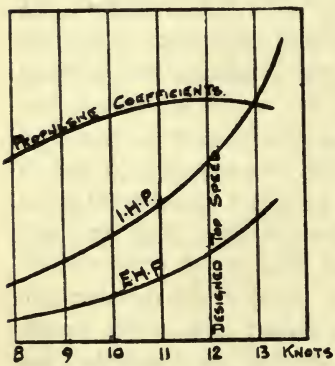


Fig. 56.

Modern single-screw cargo vessels of full form	·45 to ·5
(The former being a good figure for estimates)	
Modern twin-screw vessels of fine lines with independently worked auxiliaries (liner type)	·5 to ·55
Cross Channel steamers	·5

From the above it will be seen that the average propulsive co-efficient is ·5, showing the necessity of generating twice as much power as would be required to overcome the tow rope resistance.

Losses. Commencing at the engines, at which the I H P is obtained. the first loss encountered is initial friction, and is due to the dead-weight of working parts, working of auxiliaries off main engines, friction of packing and bearings, etc. This initial friction causes a loss to the I H P, a good average for which is about $7\frac{1}{2}$ per cent. of the I H P. It is practically constant at the top speeds of different vessels, but in the same vessel it varies directly as the speeds.

To show how it varies in the same vessel, take the following example of a vessel of 800 I H P and 10 knots speed. Suppose the speed is reduced to 5 knots, then assuming the I H P to vary as the cube of the speeds, we have :

$$800 \times (\tfrac{5}{10})^3 = 100 \text{ I H P at 5 knots.}$$

The initial friction at the top speed would consume

$$800 \times 7\tfrac{1}{2} \text{ per cent.} = 60 \text{ H P.}$$

and this varying as the speed ,we would have

$$60 \times \tfrac{5}{10} = 30 \text{ H P}$$

consumed by the initial friction at 5 knots speed. Now, while at the top speed we have $7\frac{1}{2}$ per cent. of the I H P consumed by means of initial friction, at the reduced speed of 5 knots we have 30 per cent. consumed in this way. The next loss is the load friction, it being equal to the thrust or load friction on the thrust block. The amount of this loss is about $7\frac{1}{2}$ per cent. of the I H P, this being a good average figure. Load friction varies with the speed and thrust of propeller, or practically as the I H P. We therefore see that before reaching the propeller we have lost a total of 15 per cent. of the I H P, there being now only 85 per cent. left, which is termed the "propeller horse power." The ratio of this "propeller horse power" to the I H P (85 per cent.) is known as engine efficiency and $= \dfrac{\text{P H P}}{\text{I H P}}$. Now, at the propeller we have a further loss—first of all the propeller loss itself, which is due to slip and friction of the water on the blades, and then the augmentation of resistance allowing for the wake again. Slip is the loss caused by the yielding of the water at the propeller and the screw not progressing to the full extent of its pitch; for instance, a vessel at 10 knots travels $10 \times 6{,}080 = 60{,}800$ ft. per hour with a propeller of 13 ft. pitch and 100 revolutions per minute ; $60 \times 100 = 6{,}000$

revolutions per hour, and 6,000 revolutions $\times$ 13 ft. pitch = 78,000 ft. per hour, which is the resultant forward propeller speed, while the vessel only travels 60,800 ft. in the same time, a difference of 78,000 — 60,800 = 17,200 ft., which is 22 per cent. apparent slip ;

or it may be written $\dfrac{P\,R - v}{P\,R} \times 100$ = per cent. of apparent

slip, P being the mean pitch of the screw in feet, R the revolutions per minute, and v the speed of the ship in feet per minute. In this case we would have

$$\frac{13 \times 100 - \dfrac{10 \times 6{,}080}{60}}{13 \times 100} = 22 \text{ per cent.}$$

This is not the real slip, as for this it is necessary to take into account the wake at the after end of the vessel, which tends to reduce the speed of the water past the propeller from that of the ship. However, from the above apparent slip the meaning of this propeller loss will be seen. The propeller frictional resistance is also to be accounted for in making up the propeller loss. It may be said that in efficiently designed propellers not more than 70 per cent. of the propeller horse power can be actually used in the propulsion of the vessel, and although some experiments have given better results, the 70 per cent. is a safe figure to use. Up to now, we have 85 per cent. of the I H P, equal to the propeller H P, and from this P H P we have to deduct 30 per cent. for propeller losses, since, as above mentioned. it is only possible to usefully employ 70 per cent. by means of the propeller. (85 per cent. of I H P = P H P) $\times$ 70 per cent. = 59·5 per cent. of I H P remaining.

Another loss, which is caused by the propeller, though not accounted for in propeller losses, is the augmentation of resistance. From the stream line theory we know that by the rounding in of the stream lines aft the vessel receives a pressure helping her forward motion. The propeller, disturbing this stream line action of the water at the after end, therefore causes an increase of resistance over what would be found were the propeller absent, as in the case of the tow rope resistance. It is obvious that in a single-screw ship the augment will be larger than in the case of a twin-screw vessel, as in the latter

the screws are placed further away from the ship, and, therefore, having less effect upon the stream lines ; but here we must make allowance for "wake gain" obtained by the propeller from the forward motion of the water at this part of the vessel. A single-screw ship obtains a larger advantage from the "wake" than one with twin-screws, because of the forward motion of the "wake" being greatest at the centre line. We therefore see that on account of the above mentioned augment and "wake gain," we have a deduction and also a gain to consider. The gain due to the "wake" is of less amount than the augment of resistance, and the ratio that the former is to the latter is termed "hull efficiency."

$$\frac{\text{Wake gain}}{\text{Propeller augmentation of resistance}} = \text{hull efficiency.}$$

Typical values for hull efficiency are as follows :

Single-screw vessels 90 to 95 per cent. ...	of the horse power remaining after having made the deduction for the efficiency of the propeller.
Twin-screw vessels 95 to 100 per cent. ...	

Summing up the foregoing we have :

		Per cent.
Indicated Horse Power	=	100
Deduct H P consumed in initial friction ...	=	$7\frac{1}{2}$
		$92\frac{1}{2}$
Deduct H P consumed in load friction ...	=	$7\frac{1}{2}$
Propeller Horse Power	=	85
To allow for propeller losses take only 70 per cent.	=	× ·70
	=	59·5
For hull efficiency take 95 per cent. ...	=	× ·95
Effective Horse Power remaining	=	56·5

Therefore the propulsive co-efficient in this case is :

$$\frac{\text{E H P}}{\text{I H P}} = \frac{56{\cdot}5}{100} = {\cdot}565.$$

Composition of I.H.P. The following summary shows how the total generated power is consumed :

Eddy-making Resistance.
+ Wave-making Resistance

= Residuary Resistance
+ Skin Frictional Resistance
+ Air Resistance

= Total Resistance overcome by the E H P
+ Loss due to Hull Efficiency
+ Loss due to Propeller Slip, Friction, etc.

= Propeller Horse Power
+ Loss due to Load Friction
+ Loss due to Initial Friction

= Indicated Horse Power.

Nominal Horse Power. (N H P), as given in the ship's register, is, like the registered tonnage, a fallacious figure, and one of little use in representing a close valuation of the vessel's actual power, which is the I H P. The proportion of N H P to I H P is found to have great variation when comparing different ships, an average figure being : N H P = $\frac{1}{6}$ I H P, though it varies greatly between $\frac{1}{4}$ I H P and $\frac{1}{8}$ I H P.

CHAPTER XII.

ESTIMATING HORSE POWER. THE ADMIRALTY CO-EFFICIENT METHOD, AND THE ASSUMPTIONS MADE. AVERAGE VALUES OF ADMIRALTY CO-EFFICIENT. EXTENDED LAW OF COMPARISON, MODEL EXPERIMENT METHOD, INDEPENDENT ESTIMATE AND ANOTHER METHOD. HIGH SPEEDS AND LOW SPEEDS. SPEED TRIALS. PRECAUTIONS NECESSARY. OBTAINING THE TRUE MEAN SPEED. COAL CONSUMPTION.

Admiralty Co-efficient. The most commonly used approximate method for estimating I H P is that known as the "Admiralty Co-efficient," the formula for which is as follows :

$$\text{I H P} = \frac{V^3 \times D^{\frac{2}{3}}}{C}$$

where V = the speed in knots per hour,
D = the displacement in tons,
C = the "co-efficient of performance" obtained from previous similar ships, where V, D and I H P are known, by :

$$C = \frac{V^3 \times D^{\frac{2}{3}}}{\text{I H P}}$$

The following assumptions are made :

1. That the total resistance will vary as the skin frictional resistance, which will therefore vary as the wetted surface area. Since the wetted surface area of two similar ships varies as the displacement to the two-third power ($D^{\frac{2}{3}}$), we can therefore use $D^{\frac{2}{3}}$ as representing the proportion of this resistance.

2. That the resistance will vary as the square of the speed. Of course, the 1·83 power ($V^{1 \cdot 83}$) would be more accurate for skin frictional resistance, but the difference will help to account for wave resistance. The E H P necessary to propel the vessel will vary as the cube of the speed (V^3). This will be obvious since we have assumed the resistance to vary as V^2, and we know that E H P $= \frac{R \times V}{33000}$, so therefore we use V to the third power up to the time

of obtaining E H P. The speed integer is, therefore, in all cases *one* high when dealing with power than in the case of resistance.

3. That the I H P of the engines will vary as the E H P.

It should be strictly remembered that in this method we deal with similar ships at corresponding speeds—*i.e.*, speeds which vary as the square root of the lengths of the vessels, and in such cases only can it be relied on to give satisfactory results. In estimating, the vessel chosen as the basis should be of similar form, also fairly alike in size and type as the proposed vessel. The co-efficient should be taken at the top designed speed so as to have the same engine efficiency in both cases. It is, therefore, a difficult task to obtain a vessel where the engine efficiency is the same as that of the proposed vessel at the corresponding speed, since the corresponding speed of the basis vessel may be so much different to her designed speed that the engine efficiency will have greatly altered. The importance of making comparisons only with vessels in all ways similar to each other is therefore seen. We first assumed the total resistance to be skin frictional resistance (R_s) varying as W S $\times$ V^2, and consequently that E H P will vary as W S $\times$ V^3 (W S being the area of the wetted surface), and according to the assumption that I H P varies as E H P, then I H P will also vary as W S $\times$ V^3. W S $\times$ V^3 may be considered as a measure of the I H P since we thus assume it to vary. We may term W S $\times$ V^3 as a figure having a relation to the I H P, the proportion being :

$$\frac{W\,S \times V^3}{I\,H\,P} = C$$

In similar ships this proportion C would be very nearly the same, and may, therefore, be used in proportioning the power of one ship from another by

$$\frac{W\,S \times V^3}{C} = I\,H\,P$$

In the designing and estimating stages it is improbable that the wetted surface area W S would be known, therefore W S is substituted by $D^{\frac{2}{3}}$, which has the same variation, and the formula is reduced to one which can be quickly applied, since D is one of the first figures determined at these stages.

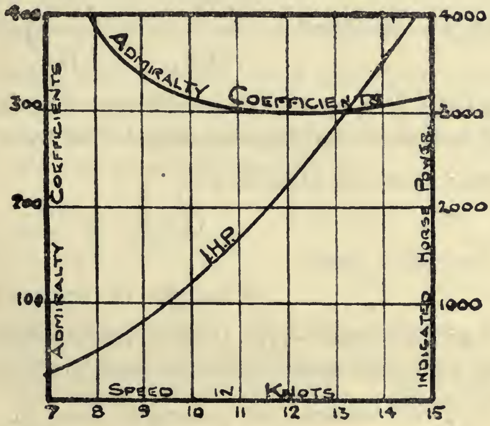
The formula, therefore, now becomes as given at the commencement.

The following are average values for C :

Atlantic liner, about 23 knots	270
Large cargo and passenger, of medium form and speed	290
Cross-Channel, about 20 knots	220
Large cargo, about 12 knots	300
Small cargo, about 9 to 10 knots	230
Trawler, about $9\frac{1}{2}$ to 10 knots	130
Herring drifter, about 9 knots	115

At different speeds of the same ship we have a variation in the value of C. In Fig. 57 the variation is shown for a vessel of ordinary proportions about 390 ft. ling and 7,800 tons displacement. The

Fig. 57.



value of C varies greatly in different vessels, but when discriminately obtained from a basis vessel the formula is very useful in the early stages of a design.

Extended Law of Comparison Method. In making a comparison between two vessels by this method it is assumed that the total resistance follows the "law of comparison" When applied to vessels of similar size and form, the results are fairly good, but, of course, it cannot be used in comparing a ship with a model, since in such a case the skin frictional resistance must be specially separately computed for the reasons mentioned and as described in the article on Resistance. With two similar vessels the difference due to assuming the skin resistance as following the "law of comparison" is

extremely small. It is also assumed that the engine efficiency at the speeds used in comparison, which should be "corresponding speeds," is equal. From the law of comparison we know that the residuary resistance varies as the length cubed or as the displacement, and also that corresponding speeds vary, as

$$\sqrt{\frac{L}{L_1}} \text{ or } \left(\frac{D}{D_1}\right)^{\frac{1}{6}}$$

therefore, assuming the I H P to vary as the E H P, we have

$$\frac{I\,H\,P}{I\,H\,P_1} = \frac{R \times V}{R_1 \times V_1} = \frac{D \times D^{\frac{1}{6}}}{D_1 \times D_1^{\frac{1}{6}}}$$

The extended law is therefore $\dfrac{I\,H\,P}{I\,H\,P_1} = \left(\dfrac{D}{D_1}\right)^{\frac{7}{6}}$

In estimating the I H P by this means, the speed of the basis ship, corresponding to that of the proposed ship, is first found as follows:

Corresponding speed of basis ship =

$$\text{Speed of proposed ship} \times \sqrt{\frac{\text{Length of basis ship}}{\text{Length of proposed ship.}}}$$

The I H P of the basis ship at this corresponding speed is next obtained from the curve of power, and multiplying this by

$$\left(\frac{\text{displacement of proposed ship}}{\text{displacement of basis ship}}\right)^{\frac{7}{6}},$$

the I H P according to the extended law is obtained.

$$\left.\begin{array}{l}\text{I H P of basis ship at}\\ \text{corresponding speed}\end{array}\right] \times \left(\frac{D}{D_1}\right)^{\frac{7}{6}} = \begin{array}{c}\text{I H P of proposed}\\ \text{similar ship.}\end{array}$$

This is a quick method of estimating I H P, and, with moderate comparisons, good results are obtained. When a smaller ship is used as the basis, the result is on the high side, since the skin frictional resistance has been taken to vary to the same extent as the residuary resistance—*i.e.*, as L^3 or D—while we know that the skin frictional resistance only varies as L^2 or $D^{\frac{2}{3}}$.

Model Experiment Method. In a previous chapter the methods of performing this experiment and obtaining the total resistance was dealt with. We know that this total resistance is to be overcome by the E H P; therefore, having obtained this amount of resistance, the E H P is found by $\frac{R \times v}{33000} = \text{E H P}$.

Where R = the total resistance in lbs. and
v = the speed in feet per minute.
or $R \times V \times \cdot 0030707 = \text{E H P}$,
where V = the speed in knots per hour.

Suppose that, from a model experiment, the total resistance of a ship at 15 knots is found to be 12,000 lbs., then

$$\text{E H P} = 12000 \times 15 \times \cdot 0030707 = 552 \cdot 73.$$

The E H P having been obtained, the I H P is next determined by using a propulsive co-efficient applicable to the type of the vessel. Say, in this case, the probable propulsive co-efficient (as obtained from the data of previous similar ships) is ·55, or 55 per cent. as it is sometimes given, then the I H P = $\frac{552 \cdot 73 \times 100}{55}$ I H P = 1050.

Independent Estimate Method. By this method we estimate the various resistances—skin frictional, eddy making and wave making, and their corresponding E H P, separately, adding them together to obtain the total E H P.

Skin frictional	...	H P
Eddy making	...	H P
Wave making	...	H P
Air resistance	...	H P (if necessary)
Total =		E H P

$$\text{Skin frictional H P} = \cdot 0030707\ (f.\text{S. } V^{n+1}).$$

It will be seen that we have the formula $f.$ S. V^{n} employed to give the resistance in lbs., which is next multiplied by ·0030707 and also by the speed so as to obtain H P, the latter being introduced by increasing the index n as shown. In the case of an ordinary ship where the index n is 1·83 we have $1 \cdot 83 + 1 = 2 \cdot 83$ to use when

estimating H P. Eddy making H P may be taken as 5 per cent. of the skin frictional H P if Froude's co-efficients have been used in the above frictional resistance formula. If Tideman's figures are used no separate estimate for eddy making H P is required, since they are high enough to allow for this. Wave making H P : This is calculated by using the formula given by Mr. Taylor, as was mentioned in the article on Resistance, but in this case the speed integer is one higher—V^5 instead of V^4—and the whole is multiplied by ·0030707, giving

$$\text{Wave making H P} = 12{\cdot}5 \times C \times \frac{D}{L^2} \times V^5 \times {\cdot}0030707$$

(The speed integer being increased when dealing with H P, for the reasons previously stated). Summing up these independently estimated H P's we obtain the total E H P, and then by using a suitable propulsive co-efficient, the I H P is found.

Another Method. The following formula gives very good results, especially when the co-efficient **a**, is obtained from a similar ship :

$$\frac{D \times V^3}{L \times \sqrt[4]{M\,S\,A} \times a} = \text{I H P}$$

D = displacement in tons. V = speed in knots. L = length. M S A = 'midship section area in square feet. **a** = a co-efficient for which the following are average values.

Atlantic Liner, about 23 knots	2·0
Large Cargo and Passenger of medium form and speed	2·4
Cross Channel, about 20 knots	2·0
Large Cargo, about 12 knots	2·7
Small Cargo, about 9-10 knots	2·5
Trawler, about 9½-10 knots	2·3
Herring Drifter, about 9 knots	2·1

High Speeds and Low Speeds. Corresponding speeds vary as the square root of the lengths of vessels, therefore we may say that speeds are relative to the length of ship. To obtain a real comparison of the speeds of various ships we should therefore compare the speeds with the square root of the respective lengths by finding the ratio :

$$\frac{V}{\sqrt{L}}$$

Say a vessel 400 ft. long has a speed of $11\frac{1}{4}$ knots,

$$\text{then } \frac{11 \cdot 25}{\sqrt{400}} = \cdot 56 = \text{speed-length ratio}$$

This ratio is representative of a cargo vessel of moderate economical speed, and is applicable to such vessels of all sizes; for instance, take a vessel 256 ft. long:

$\sqrt{256} \times \cdot 56 = 9$ knots, which is a moderate speed for a vessel of this length.

The following are average values for $\frac{V}{\sqrt{L}}$:

Cargo steamers of slow speed	·45
Cargo steamers of moderate speed	·55
Cargo steamers of good speed	·65
Cargo and passenger steamers	·75
Atlantic liners	·90
Battleships	·95
Cross-Channel steamers and cruisers	1·10
Destroyers	2·00

We may say that when the ratio is below ·5 the vessel is of low speed, at ·9 ratio the speed is high, and at 1·2 we have speeds that are excessive and only obtained with a large expenditure of power. The following example shows the application of the speed-length ratio to two vessels of the same speed but different lengths. Say, one vessel is 400 ft. long, the other 100 ft., and the speed in both cases 10 knots. In the larger vessel,

$$\frac{10}{\sqrt{400}} = \cdot 5 \text{ speed-length ratio;}$$

and the smaller vessel,

$$\frac{10}{\sqrt{100}} = 1 \cdot 0 \text{ speed-length ratio.}$$

It will be seen that while 10 knots is quite a low speed for the 400 ft. vessel, yet it is a very high speed for the vessel 100 ft. long.

Speed Trial Trips. The trial trip is one of the most eventful occasions during the vessel's career. It is the occasion during which

the owner has the opportunity of practically ascertaining the capabilities and efficiency of the engine and boilers when under full power. He also obtains the speed, maximum I H P developed by the engines, and the consumption of fuel necessary to obtain the speed and power. On the other hand, the shipbuilder takes the opportunity to obtain data for future designing purposes in the shape of the amount of I H P necessary to propel the vessel at various speeds ranging up to the maximum, number of revolutions, slip and efficiency of propellers. The following are the methods usually adopted for obtaining the speed of the vessel :

1. Successive runs in opposite directions on a measured mile.

2. A continuous run at sea, the number of revolutions being counted during the time occupied and the mean speed for the run found as afterwards described.

3. A continuous run at sea past a series of stations of known distances apart, the times being recorded when the ship passes.

4. Patent logs.

The last method is of little use for trial trips, the results not being sufficiently accurate for this purpose, although it is extremely useful for ordinary navigation.

Progressive Trials on the "Measured Mile." So as to obtain a true course for these trials, posts are erected on the coast is positions as shown in Fig. 58, which represents the measured mile near the mouth of the Tyne. The course is marked by buoys or by a compass bearing, as in the case of the Tyne mile, where the course is north and south. A number of runs are usually made over the course at various speeds, making at least one run each way for each speed. If the trial is not to be a *progressive* one, and the vessel is to be run at full power only, then two or three runs are made in each direction making four or six in all. In progressive trials it is best to make these numbers of runs for each speed taken and to find the " mean " as is afterwards described, although the mean of two runs, one with and one against the tide, would give a fairly accurate result. Some progressive trials are commenced at the top speed, while in others the first run is made at a very low speed. For instance, if the latter is adopted in the case of a vessel of 5,000 maximum I H P, the first series of runs would be made with an I H P of about 1,000, and after having carefully noted all the results, as mentioned later, the power would be increased to about 2,000, when the second series of runs would be made. The next runs would be made at progressive I H P's,

until the maximum is reached. The results of one series of runs would be ascertained as follows: The times of passing the posts carefully taken with chronometer stop watches by observers on deck, the number of revolutions and the I H P being noted by the engine-room officials. Under precisely similar conditions, these particulars are again taken with the vessel steaming in the opposite direction. The power being increased for another series of runs, records are again made. The results of progressive trials are arranged in diagrammatic form, as shown in Fig. 59. The amounts of I H P are obtained by the engine-room staff, by means of taking indicator diagrams and then setting off the results at their respective speeds;

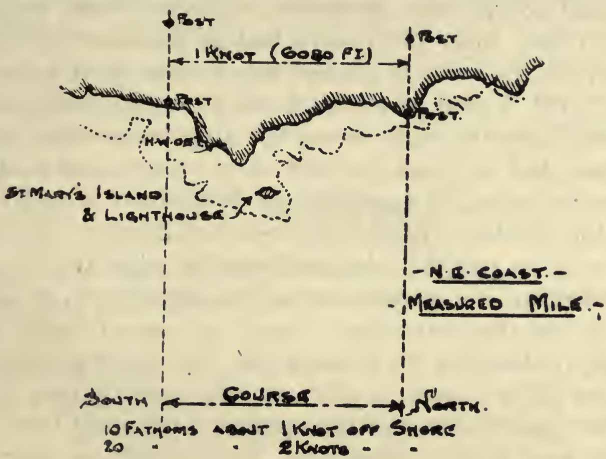


Fig. 58.

the curve is drawn as shown. The numbers of revolutions being noted for these speeds they are also set off, as shown. The Admiralty co-efficients, corresponding to the trial results, are generally calculated and a curve also drawn for them. These curves are extremely useful in the designing of future ships; for instance suppose it was required to know the amount of I H P necessary to drive a proposed similar ship at a given speed, the Admiralty co-efficient method being employed. In the above, it was said that, when using this method it is necessary to obtain a co-efficient for a previous similar ship at the corresponding speed. From the curve shown in Fig. 59, the co-efficient can be quickly read off after determining the corresponding speed, and by use of this co-efficient the I H P may be estimated for the proposed vessel. Other curves may be added to this diagram, such as slip, indicated thrust, etc.

Precautions necessary. Seeing that trial trips are always looked upon as being obtained under the most favourable conditions, it is of the utmost importance that such shall be the case. The vessel's bottom should be in clean condition and the wind and sea favourable, the machinery in perfect order and under good control. The course chosen should be of sufficient depth to suit the speed of the vessel, and not to interfere with the resistance, as is sometimes the case of high speed ships in shallow water, when the resistance is increased, although, in extreme cases of high speed vessels, very shallow water has been found to aid the speed. The resistance is influenced by means of the shallow water affecting the wave formation. It is found that normal wave formation is obtained where the depth of water, in feet, under the vessel's bottom exceeds $\cdot 28\ V^2$, V being the speed of the vessel in knots. Say a vessel of 14 knots : $\cdot 28 \times 14^2 = 55$ ft. A good steersman should be at the wheel so as to get the vessel upon the right course and keep her steady. It is very important that the vessel be kept on a straight course, this being very obvious when we remember the definition of a straight line as being the shortest distance between two points. Although the course may be straight, care must also be taken that it is in the right direction—*i.e.*, parallel to the two outer posts, or north and south on the above-mentioned course, because any other direction but this will lengthen the distance run. At each end of the course there should be a space in which to take a steady turn to get the vessel into position again and be running at full speed before passing the first post on the return run. There should be three or four chronometer stop watches used in the taking of the times, these, of course, being in perfect order. The observers should accurately start and stop the watches, and the mean of the times should be taken, because one person may anticipate, while another may delay until he sees the posts actually open. The timing is one of, if not the, most important observation of the trial trip ; a difference of one second on the mile would mean ·1 of a knot at a speed of about 20 knots per hour.

To Obtain the True Mean Speed. This is done by finding the *mean of means*, by which method the tidal and wind effects are eliminated. Since the tide varies in speed and direction of flow, it is necessary to obtain the mean speed, as seen in the following example which shows the ordinary method of finding the *mean of means* :

	Knots.					
Run 1. With tide ...	15·000					
		14·344				
Run 2. Against ,, ...	13·688		14·375			
		14.407		14·431		
Run 3. With ,, ...	15·126		14·487		14·467	
		14·567		14·503		14·472 knts mean speed
Run 4. Against ,, ...	14·008		14·520		14·477	
		14·473		14·450		
Run 5. With ,, ...	14·938		14·380			
		14·287				
Run 6. Against ,, ...	13·636					

86·396 ÷ 6 = 14·399 knots.

It will be seen that if the ordinary average of the six speeds is taken we have 14·399 knots against 14·472 as the true mean, a difference of nearly ·1 of a knot. To be strictly correct, the mean speed should be found by first of all finding the *mean time* instead of the *mean knots per hour*, as is usual, and as is done in the above example. For instance, if the above were correct, we would say that, in the case of a vessel which runs one way on the mile at the rate of 20 knots per hour and returns at the rate of 10 knots per hour, the mean speed was

$$\left.\begin{matrix}20\\10\end{matrix}\right\} = 15 \text{ knots per hour.}$$

such is not the case, as the following will show:

at the rate of 20 knots, the time occupied was	3	minutes.
,, ,, 10 ,, ,, ,,	6	,,
therefore 2 knots were run in	9	,,
being a mean of 1 knot in	$4\frac{1}{2}$	,,

$$60 \text{ minutes} \div 4\tfrac{1}{2} = 13\tfrac{1}{3} \text{ knots per hour,}$$

which was the mean speed of the vessel, and not 15 knots. This example is, of course, exaggerated in the amount of difference between the speeds (20 and 10 knots), yet it proves that the correct method is to find the *mean of the times* and then for the mean speed to take that corresponding to the mean time. In ordinary trials, however, the effect due to this is very slight. If in the above example we had found the *mean of means* for the time occupied in running the mile, the corresponding speed would have been 14·46 knots, a difference of only ·012 knot. During the running of these trials, the displacement and trim should be as near as possible to those designed for. The second method of conducting trials is performed as follows: The revolutions are counted during a continuous run at sea of 8 hours, say. and the total divided by the time occupied in minutes will give the mean number of revolutions per minute.

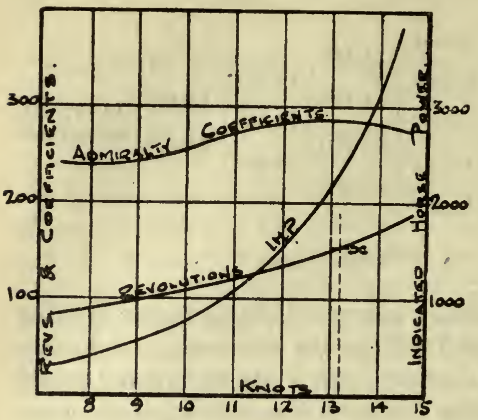


Fig. 59.

Having first of all constructed a diagram as shown by Fig. 59, from the results obtained on the measured mile trial, the speed corresponding to this mean number of revolutions is easily found. From the point on the curve of revolutions corresponding to the number found as above (in Fig. 59 the point is shown by x), drop a perpendicular down into the scale of speed, where the mean speed during the continuous run will be obtained. In Fig. 59 it is shown to be 13·2 knots.

The third method—The series of stations is obtained by placing vessels at distances apart of about three knots. Persons aboard the station ships note the times that the trial ship passes in addition to those on board the trial ship also noting the times, a good check being thereby obtained. The observers on board the station ships also make measurements of the speeds of the tide and wind, and then, by making these corrections, a most accurate result of the trial is obtained.

Coal Consumption. When the coal consumption is measured, the method employed is usually as follows: First, by using two bunkers, *a spare* one and a *sealed* one. Coal from the spare bunker is used until the vessel actually enters upon the trial, and then, with ordinary fires, the spare bunker is sealed up and the sealed bunker broken. At the end of the trial, the fires being left in ordinary condition, the latter bunker is again fastened up and the coal from the spare bunker used. By knowing the original weight in the sealed bunker, and then measuring or weighing the remaining amount, the exact quantity used during the trial can be obtained. The second method is to have a weighing party on board, and by means of weighing machines to weigh the exact amount of coal used in the fires. The amount of coal used per I H P developed during the time can then be found.

CHAPTER XIII.

THE STEERING OF SHIPS. TYPES OF RUDDER. CAUSES OF PRESSURE ON RUDDER. CALCULATION FOR PRESSURE. TWISTING MOMENT ON RUDDER HEAD. CALCULATING THE REQUIRED DIAMETER OF RUDDER HEAD. TURNING TRIALS.

The Steering of Ships. In Fig. 60 we have represented the two types of rudders usually fitted, **A** being the "ordinary" type hung at its forward edge, and **B** shows the "balanced" type, which has an amount of area forward of, as well as abaft of, its axis. In the latter it will be seen that the after stern-post is omitted, the

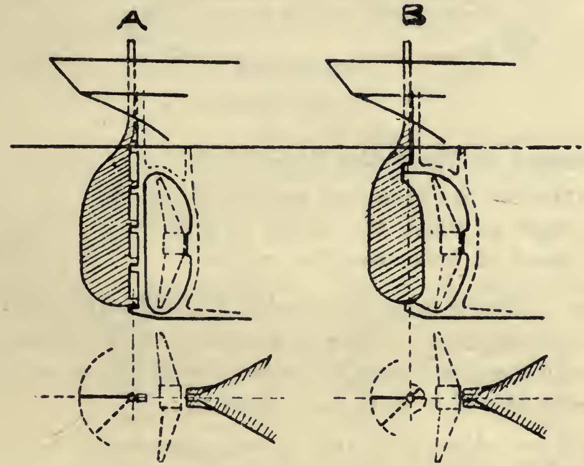


Fig. 60.

rudder being supported in only two places; sometimes, however, the bottom bearing is dispensed with and only one large bearing is fitted at the top. It is obvious that to obtain steerage effect from the rudder we must have motion of the ship through the water, or a flow of water past the rudder, so that an excess of pressure may be obtained on one side of the rudder, which will then cause the vessel to alter her course. In the case of a screw ship the propeller race is, in this way, the means of giving the vessel steerage way, even before the vessel herself has obtained motion. To keep a single-screw vessel on a straight course it is usually found necessary to hold the

rudder over to a small angle, but for our present purpose we will assume that the vessel's rudder, while on the centre line, will give this result. Suppose a vessel, while steaming in a forward direction, is put under a port helm—*i.e.*, the rudder is put over to the starboard side, as shown in Fig. 61. This obviously causes an excess of pressure (**P**) on the starboard side, forcing the vessel's stern in an outward direction (to port), as shown by the arrow **D**, resulting in the vessel changing her course and her head coming round in a starboard direction. The steering power of a vessel being firstly dependent upon this pressure **P**, it will, therefore, be interesting to investigate the items influencing its amount.

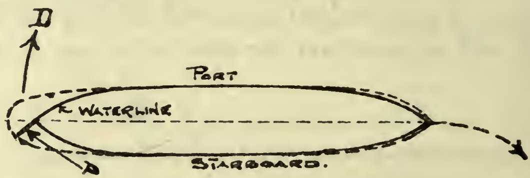


Fig. 61.

The pressure on the rudder depends on:

The area of the immersed surface of rudder.
The angle at which the rudder is held over.
The speed of water past the rudder.

While the rudder is lying in a fore and aft direction, according to the above assumption, we have no pressure; but if it is held at right angles to the line of motion of the water a direct resistance is obtained, the pressure of which is, in lbs., equal to:

$1{\cdot}12\ A\ v^2$, where A = area of immersed surface in square feet and v = the speed in feet per second.

If the line of motion is assumed to be parallel to the keel line, then at any angle to this line the above pressure varies as the sine of the angle:

$$1{\cdot}12\ A\ v^2 \sin\theta = \text{pressure in lbs. for any angle } \theta.$$

The line of motion is not, however, parallel to the keel line, and it may be fairly assumed to be parallel to a water-line representing a mean of those contained in the immersed portion of the after part of the vessel. The angle used in the formula should, therefore, be increased to suit this. This point is illustrated in Fig. 61, where the mean water-line is shown, and it will be seen that while the rudder is only

held over to 45 deg. (usually the most efficient), yet the impingement of the water, giving the resultant pressure **P,** is acting nearly at right angles to the rudder surface. The speed used should be that of the water past the rudder, and here the effect of the propeller must be accounted for by increasing the vessel's speed by 10 per cent., which will allow for the difference between the vessel's speed and propeller speed, also accounting for the forward motion of the wake.

Twisting Moment on Rudder Head. When a rudder is held over by means of steering gear or tiller and subjected to such a pressure as the above, a twisting stress is brought to bear upon the rudder stock. If the centre of pressure is found relative to the axis of the rudder we then have the lever through which the above pressure is acting, and the pressure in lbs., when multiplied by the lever in feet, will give the twisting moment in foot-lbs.

$$\text{Twisting moment in ft.-lbs.} = (1{\cdot}12\ A\ v^2 \sin\ \theta) \times l$$

l representing the lever.

At first sight it may appear that the centre of pressure will coincide with the centre of gravity of the immersed rudder area ; this is not the case, and for rectangular plates we may take the distance from the leading edge as follows :

10 deg.	= ·24 of breadth.	50 deg.	= ·42 of breadth.
20 ,,	= ·315 ,,	60 ,,	= ·44 ,,
30 ,,	= ·365 ,,	70 ,,	= ·46 ,,
40 ,,	= ·400 ,,	80 ,,	= ·48 ,,
45 ,,	= ·405 ,,	90 ,,	= ·50 ,,

Owing to varying speed and direction of steam lines at the stern of a ship and the interruption caused by the propeller, the above values may not be exactly correct when dealing with a ship's rudder, but they may be considered as being fairly near to the correct figure. When a vessel is moving in a sternward direction the after edge of the rudder becomes the leading edge, and the centre of pressure should then be found from this edge. Its position will now be further from the axis of the rudder and the force **P** will be now acting through a larger leverage than when the vesesl was moving in a forward direction. The force **P** is not, however, so large when going astern as when ahead, on account of the speed astern being much less than that ahead. It is usual to take speed astern as being

equal to three-quarter speed ahead. The reduced speed, lessening the rudder pressure, has a greater effect than the increasing of the lever in the case of ordinary rudders; therefore, for this type, the stress when going astern is less than when ahead; but in the case of balanced rudders it is generally found that the reduced speed has a smaller effect than the large increase of lever found in this type, and, therefore, the maximum stress is obtained when moving sternwards, and for balanced rudders the calculations should be made corresponding to this condition. In the case of figure 62 (**A**) the lever **y** would be used instead of **x**. To find the centre of pressure of an actual rudder: Take, for instance, the rudder shown in Fig. 62 (**B**), and upon it construct a rectangle having the same area and overall width. Now obtain the centre of gravity **G** of the actual shape of the rudder surface and the centre of gravity G_1 of the squared-up surface. Next find the centre of pressure **p** of the squared-up surface by multiplying the overall width by the co-efficient corresponding to the angle as given above; say 10 deg. = W × ·24 = *d*, giving the point **p**. Since the centre of gravity of the actual

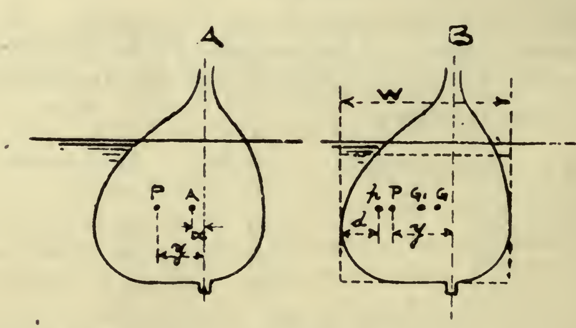


Fig. 62.

surface was found to be forward of the centre of gravity of the squared-up surface, the actual centre of pressure will also be forward of **p**, which corresponds to the squared-up surface. The amount of correction for this is dependent upon the distance **G** G_1, and if to the distance **d** we add **G** G_1 × ·24 (for 10 deg.) the point **P** will be obtained, which is very near to the correct position of the centre of pressure of the rudder surface at 10 deg. inclination to the line of the motion of the water. It will be noticed that the above applies to a vessel with balanced rudder and moving sternwards. The twisting moment in inch-tons will be equal to:

$$(1{\cdot}12 \text{ A } v^2 \sin \theta) \times l \times \frac{12}{2240} = \text{T}.$$

And from this the necessary diameter of rudder-head can be found by means of the following formula :

$$d = \sqrt[3]{\text{T} \times e},$$

where d = the diameter of rudder head, in inches, and e = a co-efficient varying according to the material.

For cast steel	$e = 1{\cdot}02$
For wrought iron	$e = 1{\cdot}28$
For phosphor bronze	$e = 1{\cdot}70$

Turning Trials. Fig. 63 shows the course taken by a vessel when steaming ahead with rudder hard over and turning a circle, the position of the vessel being shown at various points. The outer line, which is swept out by the vessel's stern, is the boundary within which the vessel can turn ; the middle line is the one which the vessel's centre of gravity is traversing, and the inner line is the one upon which the pivoting point of the vessel lies, and is tangent to the vessel's centre line (*see* enlarged sketch Fig. 64). To make observations for these trials a person is situated at the aft end, and another at the fore end of the vessel, on the centre line, as **M** and **N** in Figs. 63 and 64, their distance apart being known. **O** represents a buoy which is moored, and around which the vessel is making revolutions, the above observers simultaneously sighting the angles **m** and **n** at times of which they are acquainted by the blowing of the vessel's whistle. A triangle **M N O** (*see* Figs. 63 and 64) can then be constructed, and the distance of the vessel from the buoy can then be found by placing the apex upon **O**, the position of the buoy. A number of observations are made, and a third observer having noted the bearing of some fixed object

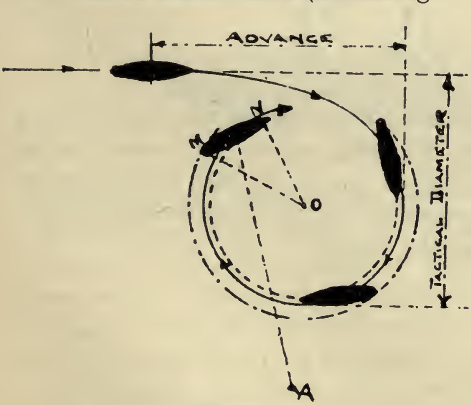


Fig. 63.

on shore, such as A in Fig. 63, or the compass bearing at the times, the exact positions of the vessel at the various times can be shown in the diagram, and the path of the vessel can be drawn, as shown in Fig. 63. In Figs. 63 and 64 it will be seen that the vessel's bow points inside the tangent to the path of the vessel's centre of gravity **G**. The amount of this is known as the drift angle. If a line **O P** is constructed at right angles to the vessel's centre line, so as to cut

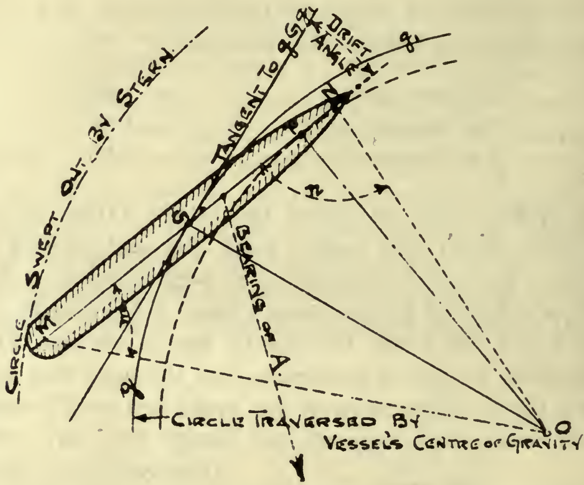


Fig. 64.

the position **O** of the buoy, the point **P** is obtained, and this is known as the pivoting point, being the point where the vessel's centre line is tangent to a circle (the inner one in Fig. 63), and this point **P** is the one about which the vessel is turning, being usually situated near the extreme fore end of the vessel.

CHAPTER XIV.

THE STRENGTH OF SHIPS. THE POSITIVE AND NEGATIVE LOADING OF SHIPS, AND STRAINS CAUSED THEREBY. CONSTRUCTUON OF THE CURVES OF WEIGHT, BUOYANCY, LOADS, SHEARING FORCES AND BENDING MOMENTS. FORMULA FOR THE DETERMINATION OF THE AMOUNT OF STRESS. EFFECT OF SUPERSTRUCTURES.

The Positive and Negative Loading of Ships and Strains Caused Thereby. Throughout the length of a ship we have a most irregular distribution of weight, obtaining support from the surrounding buoyancy which is varying somewhat uniformly all fore and aft. The effect of the so distributed forces of weight and support is to cause alternate excesses of positive and negative loading which set up stresses on the vessel's structure. The nature of these excesses vary between the light and load draughts; take the vessel shown in Fig. 65 for instance. If we imagine the vessel to be severed at the end of each compartment, each portion being water-tight and allowed to freely take up its independent position, we would find some to

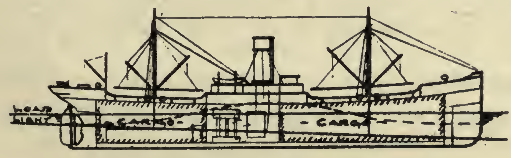


Fig. 65

sink further and others to rise, while they would all, perhaps, alter their trim (*see* Figs. 66 and 57). The irregularities of weight and buoyancy can be best illustrated by means of curves showing their distribution. The curve of weight is first drawn, the construction of such a curve for the load condition being shown in Fig. 68, where it will be seen that each item is taken separately, and built up one upon another. A vertical scale having been decided, such as $\frac{1}{4}$ in. $= 1$ ton, the weight of each item is calculated per 1 ft. of its length, this amount being set off over the corresponding length covered. The various items having been set down, the curve produced will be extremely irregular, as shown in Fig. 68, but the smaller irregularities may be discarded and the curve of weight drawn, as shown in Fig. 69. The curve of buoyancy is next drawn, the amount of buoyancy per foot length being found at various positions of the

vessel's length, and set off to the same scale as the curve of weights, as shown in Fig. 69. We now have a graphic representation of the distribution of the two forces, weight and buoyancy. The areas of the two curves must be the same, as must also be the

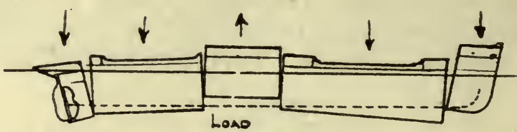


Fig. 66.

longitudinal position of the centres of gravity of the areas, to conform with the conditions of stable equilibrium. From these two curves we can now find the resultant effect upon the vessel's structure, as follows : The curve of *loads* is next constructed, as shown in Fig. 70, this curve representing the differences between the weight and buoyancy ; the positive loading, or excess of weight, being shown below the base line **A B,** and the negative loading, or excess of buoyancy, being shown above the line. The next curve constructed is the *shearing forces.* Commencing at the end **A** of the curve of *loads*, the area, shown shaded, up to **X** is found, this area being then converted into tons shearing force in the following way :

Say the longitudinal scale is : $\frac{1}{8}$ in. = 1 ft. $\therefore$ 1 in. = 8 ft.

,, vertical ,, $\frac{1}{4}$ in. = 1 ton $\therefore$ 1 in. = 4 tons.

Then 1 square inch = 32 tons.

If the area up to 9 is three square inches, then $3 \times 32 = 96$ tons of shearing force at this point, which is set off as an ordinate at **X** (*see* Fig. 70). The areas of the *loads* having been found up to various points and converted as above, the curve of *shearing forces* can be drawn in through the offsets. The curve ascends until the termination of the positive loading at **C,** after which, on account of the *load* being here above the line **A B,** and, therefore, minus quantities, the curve then descends. It crosses the base line at **D,** when the amount of negative loading has counterbalanced the positive loading, and goes on descending until the positive loading is again in excess after **E** is reached, when the ascension ultimately brings the curve back to the base line at **B.** From the curve of *shearing forces* the curve of *bending moments* can be obtained in exactly the same

way as the *shearing forces* were obtained from the *loads*, again allowing for the difference in the scales. It will be seen that the *bending moments* reach their maximum where the *shearing forces* cross the base line. The bending moment is, in this case, one tending to "hog" the vessel because of the excess of weight at the ends and the buoyancy excess at 'midships. When without cargo a "sagging" moment would most probably be found on account of an excess of weight 'midships and buoyancy excess at the ends. The above refers to a vessel floating at rest in still water. If we now consider the vessel placed amongst a series of waves, the bending moments will be found to be greatly increased. It is usual to

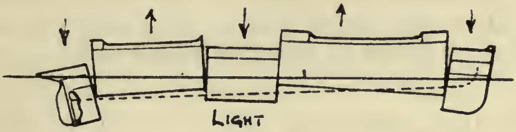


Fig. 67.

suppose the vessel poised upon the crest of a wave whose length from trough to trough is equal to the length of the ship, and the wave height from trough to crest to be $\frac{1}{20}$th of its length when 300 ft. long and below, and $\frac{1}{25}$th when above that length. The wave profile is trochoidal in shape, and its position on the vessel is such as will produce the same displacement and longitudinal centre of buoyancy as in the still water condition. The dotted line in Fig. 65 shows the wave profile, while the dotted line in Fig. 69 shows the resultant buoyancy curve. From the latter figure it will be seen that a deduction of buoyancy has been made at the ends and an addition at 'midships, thereby increasing the amounts of positive and negative loading, as given by the curve of *loads*, the further

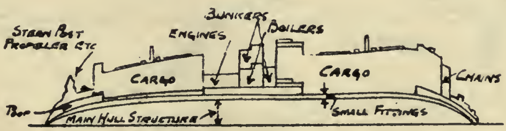


Fig. 68.

result being a largely augmented "hogging," bending moment. When considering the vessel with the wave crest 'midships, it is usual to assume 'midship bunkers and feed-tanks empty, which, consequently, tends to increase the "hogging" effect, and for the trough 'midships to take them as being full, this obviously tending

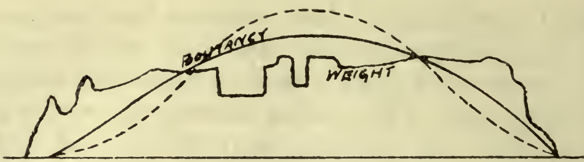


Fig. 69.

to increase the "sagging," bending moment. The bending moments given by the curve are in foot-tons, and from these amounts the resultant stress upon the vessel's structure can be ascertained when used in the following formula :

$$\frac{M \times Y}{I} = \text{Stress in tons per square inch,}$$

where M = the bending moment.
Y = the distance from the neutral axis to the point at which the amount of stress is required.
I = the moment of inertia of the cross-section about the neutral axis.

The moment of inertia of the vessel's cross-section is next to be found. In this calculation only the longitudinal members of the structure are taken into account (*see* Fig. 71). The neutral axis **N A**, which passes through the centre of gravity of the cross-section, is also found, and the moment of inertia is one taken about this axis. The greatest stress will be obtained where **Y** is greatest ;

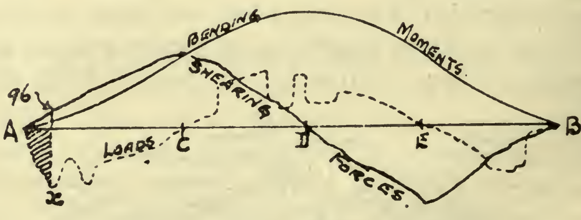


Fig. 70.

for instance, if we suppose a "hogging" stress applied to the vessel represented by Fig. 71, **Y** is greatest when taken to the top of the upper deck sheerstrake. This formula is applied to the vessel at points which appear to be weakest, or where the bending moment is largest, and the stress upon the points in question being so found, the strength of the vessel is known. The greatest stress is nearly

in all cases found when the vessel is poised upon a wave crest, and, from experience, it is found that the figure for this stress which will produce a vessel free from weakness varies with the size of the ship. In small ships of the coasting types, 2 tons is a safe figure, while for medium-sized vessels of mild steel construction, about 6 tons in tension, and 5 tons in compression are suitable. In very large ships the figures may be further exceeded; for instance, in the Cunard Company's *Lusitania*, while enduring a hogging stress with a wave crest at 'midships, the corresponding stress is given as 10·6 tons tension on the shelter deck and 7·8 tons compression on the keel.

In computing the moment of inertia, all longitudinal portions of the structure, covering, at least, half of the vessel's length, should be taken into account, allowance being made for all openings that would cause a weakening. For the parts in tension, a deduction should be made for rivet holes by taking only $\frac{9}{11}$ths of the area of such material instead of the full amount. The principle adopted in finding the M I is as follows: Suppose the shaded portion in Fig. 72 represents the section of the sheerstrake plate of a vessel, its area being equal to **A**. If I = the total amount of inertia of the plate about the neutral axis **N A**
and I_x = the moment of inertia of the area about its own axis **m n**, which amount is equal to $\dfrac{A \times d^2}{12}$

then $I = (A \times h^2) + I_x$.

It will be seen from the above that to reduce stress it is necessary to increase the moment of inertia, since this is the denominator of the formula $\dfrac{M \times Y}{I}$ which gives the stress, a large moment of inertia, therefore, giving a small stress. To obtain a large M I, the material farthest away from the neutral axis should be made of the heaviest scantling, because its area is multiplied by the square of its distance from that axis, viz.: $(A \times h^2)$. This shows the advantage of placing material as

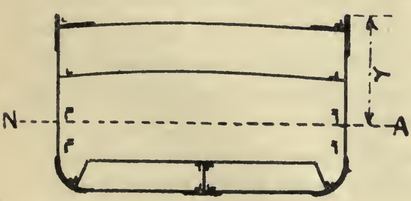


Fig. 71.

far away from the neutral axis as possible; but this should only be done when the added material will stand the large stress that comes upon it when placed at a large distance from the N A. At first thought one is apt to think that high superstructures would be the means of affording further strength to a vessel, because of largely increasing the moment of inertia; but when these are only lightly constructed, such as casings, deckhouses, boat decks, etc., it is found that the increased length of Y has a greater effect than the increased I, and the stress given by $\frac{M \times Y}{I}$ is consequently larger than when they were unaccounted for. The light scantlings of such erections would not stand the increased stress; therefore, they must be arranged so that they will take no part in the longitudinal bending of the vessel, and the vessel's longitudinal strength be quite independent of them. This is done by fitting an expansion joint, which is merely an overlap without any riveted connection, being, therefore, able to work freely during any bending of the vessel, and thereby preventing fracture, which may otherwise occur.

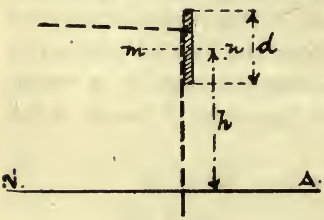


Fig. 72.

CHAPTER XV.

FREEBOARD AND REASONS OF ITS POSITION. RESERVE OF BUOYANCY. EFFECT ON STABILITY AND STRENGTH. BOARD OF TRADE RULES. AMENDMENTS IN 1906.

Freeboard, and Reasons of its Provision. Roughly speaking. freeboard may be defined as being the height of the upper deck at side, amidships, above the load water-line. The purposes for which it is provided are as follows : 1. To provide a reserve of buoyancy as a margin against leakage or accident, and to give lifting power when in a seaway. 2. Sufficient height of deck above water so as to obtain a safe working platform for those on board, and to give immunity from danger to deck fittings, etc. 3. Stability. 4. A depth of loading so as to prevent excessive strain being brought upon the vessel's structure. We, therefore, see that buoyancy, stability, structural strength and freeboard are subjects that must be closely related to each other, the latter being determined according to the qualities of the other three.

Reserve of Buoyancy. This is the volume of a ship which is not immersed and is water-tight. It includes, in addition to the upper portion of the hull, any erections, such as poop; bridge, forecastle or raised quarter deck, that have efficient water-tight bulkheads at their ends. It is possible for a vessel to float with her deck level with the water, so as to have practically no reserve buoyancy whatever, but in such a condition she would have no rising force and, when in a seaway, every wave would simply wash along or over the deck, undoubtedly carrying away all deck structures, and probably eventually causing the foundering of the vessel. In the case of a vessel which has a reserve of buoyancy and on a wave with the crest about amidships, we have, for a moment, an excess of displacement, and the vessel immediately tends to rise so that her weight may no longer be exceeded by the displacement of water. Without the reserve of buoyancy, the vessel would not possess the power of heaving and, as in the above case, would be submerged with every succeeding wave. The second reason is a most obvious one, so as to allow of the vessel being navigated with a minimum of personal risk.

Effect on Stability. A vessel with a good freeboard is able to incline to a much larger angle before the deck corner reaches the water than one whose freeboard is small. This has the important effect of pulling out the centre of buoyancy and increasing the righting levers on account of increasing the value of the equation

$$\frac{v \times hh_1}{V}$$

which forms part of the formula that gives the righting lever (*see* Chapter VIII. and Fig. 40). In Chapter VIII. the influence of freeboard on a vessel's stability was shown in Fig. 45, where the increased range resulted in the new curve G_2 Z_2.

Strength. In the previous article we dealt with the causes of longitudinal bending moments in a vessel; let us now assume that we have a bending moment which, from experience and such calculations, we know will allow of a safe margin of safety with the vessel in a load condition. Now, consider the effect of overloading her; The extra amount of cargo that has to be put on board cannot very well be placed near amidships on account of the machinery space, and is, therefore, placed in the holds or upon the deck. The vessel is now immersed further into the water, the effect being to increase the amount of supporting buoyancy mostly in the vicinity of amidships, this being the fullest portion of the vessel. The weights would increase nearer the ends, and the combined effect of the extra buoyancy and weight, upon a curve of loads, will obviously have the resultant effect of increasing the bending moment and, consequently producing an increased stress.

Board of Trade Rules. The rules for the computation of freeboard are arranged for four types of ships, a corresponding table of freeboards being given for each. Table A is for first-class cargo-carrying iron or steel steamers not having spar or awning decks—*i.e.*, ships of the full strength. This table corresponds to flush-deck vessels, but when substantial erections are fitted, allowances are made for their contribution to the vessel's reserve buoyancy. Table B is for cargo-carrying spar-deck vessels, which type is of a lighter construction than the Table A vessel, the freeboards being increased on that account. Table C is for cargo-carrying awning-deck vessels, which is a still lighter type of construction, having a greater increase in freeboard. Table D is for sailing vessels. The tables are drawn up with columns in which is given the freeboard for vessels of certain

depth and proportionate length, a correction being necessary for a difference in length to that specified in the tables. The freeboards given for the proportionate dimensions also vary according to the fineness of the vessel's form, which is represented by a co-efficient, the fuller vessels having the largest freeboard. In Fig. 73 a diagram is shown given the variation of freeboard (full lines) on a basis of length of ship with proportionate table depth. The four tables are

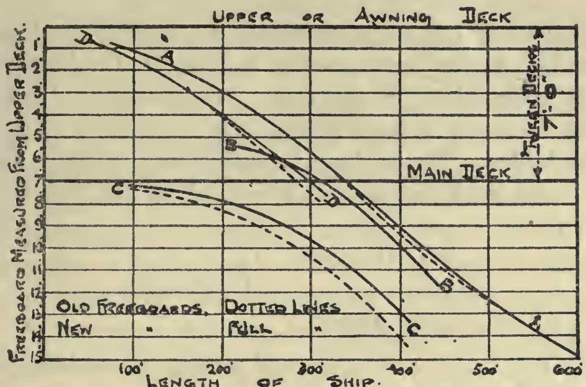


Fig. 73.

represented for typical cargo-carrying vessels, A, B and C being for steamers of a table co-efficient of ·8, while D is for iron sailing vessels of ·74. (This co-efficient is one specially estimated for freeboard purposes, but is usually about equal to the block co-efficient of displacement.) The reason for the difference in freeboards in A, B and C is because of the lighter scantlings of the latter two; the addition of freeboard in their cases reducing the weight carried, which, therefore, tends to make the stress in all cases about equal, the lighter vessel carrying the smallest amount. Fig. 74 shows another comparison of the four tables, giving the position of the respective load-lines for a vessel 300 ft. long, and 25 ft. depth, moulded, to the uppermost deck, the co-efficient being ·74 in all cases. The relation of displacements is near to the following:

Displacement of B is about 96 per cent. of A.
,, ,, C ,, 88 ,, ,,
,, ,, D ,, 97 ,, ,,

In the above comparisons it must not be forgotten, however, that the Table A vessel is one of the most meagre description possible—

i.e., she is taken as being just equal to the bare table rule. Under ordinary average circumstances in a vessel of this size and type, say with a poop, bridge and forecastle covering about half of the length of the vessel, we would have the following deductions to make :

For	Excess of sheer	about	7 ins.
,,	Erections	,,	$9\frac{1}{4}$,,
,,	"Iron Deck"	,,	$1\frac{3}{4}$,,

making a total of 1 ft. 6 in. reduction in the freeboard and giving the water-line $\mathbf{A}_1$, as shown dotted in Fig. 74, the increase of displacement being about 8 per cent. In estimating freeboard the first thing necessary is to determine the co-efficient of fineness for use in the tables. This is done by using the registered dimensions and the under deck tonnage as measured by the Board of Trade surveyors :

$$\frac{\text{U D Tonnage}}{\text{Reg. Dims.}} = \text{Co-efficient.}$$

To obtain this accurately many corrections are necessary to the dimensions and tonnage in almost every case ; but it should be remembered that the co-efficient sought is one the vessel would have if framed on the ordinary transverse system—*i.e.*, with ordinary

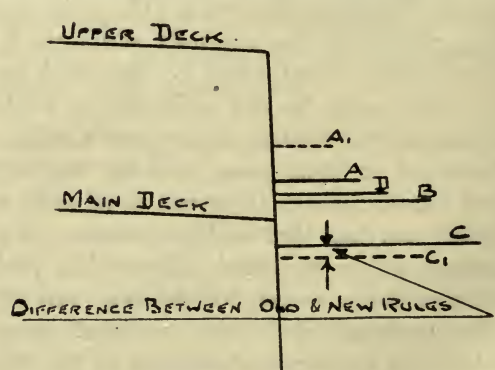


Fig. 74.

floors (no double bottom) and frames of Lloyd's 1885 rule scantlings, also if fitted with hold and spar ceiling ; therefore, when any departure is made from this, by fitting double-bottom, deep framing, or the omission of ceiling, corrections should be accordingly made. If the

mean sheer is in excess or deficient of the rule standard, the registered depth to be used for ascertaining the co-efficient is to be increased or reduced respectively by one-third of the difference between the standard and actual mean sheer of the vessel. The freeboard can now be determinee for the table length according to the depth, moulded, and co-efficient. The correction for sheer is next made, its amount being one-fourth of the difference between the rule and the actual mean sheer of the vessel, being deducted from the freeboard for an excess in some cases, and where there is a deficiency it is added. The correction of length is now accounted for, the amounts being giving in the tables for every 10 ft. of difference between the table and actual lengths. It is added when the vessel's length exceeds that given in the table and deducted when it is less. An allowance for deck erections is to be made according to the nature of erections, and based upon the proportion that the combined length of erections bears to the length of the vessel. It is assumed that an awning-deck vessel is simply a Table A ship, with an erection extending all fore and aft, and the difference between the two table freeboards, after making the above corrections, is the allowance for the complete erection. The deck erection allowance is then based on this difference and according to the extent of the length of the ship they cover, although the allowances are actually less than the proportion of the difference between Tables A and C that the proportion of length covered would give. For instance, a poop, bridge and forecastle covering half length would obtain an allowance of 32 per cent. of the difference, the rèduction from 50 per cent. allowing for the breaks in the strength caused by the three separate erections, other corrections being made if required for "iron deck," extra strength, round of beam, fall in sheer, non-compliance with rules regarding freeing port area, satisfactory arrangements for crew getting backwards and forwards from their quarters, etc., deduction for summer voyages, and height of statutory deck-line, as stipulated in the rules, the freeboard is eventually obtained. In the case of raised quarter-deck ships, corrections are to be made in cases where the height of raised quarter-deck above main deck is less than that required by the rules, and also when the engine and boiler openings are not covered by a bridgehouse.

The present opportunity may be taken to make mention of the 1906 amendments to the tables of freeboard, which then caused no small amount of controversy from various points of view. One or

two alterations were made to the rules governing the amounts of the various corrections for co-efficient length and erections, but the most important changes were made in the tabulated freeboards. Fig. 73 is utilised to show these, the full lines representing the present-day curves, and those dotted are according to the tables existing previous to March, 1906. In the case of Table A, it will be seen that a reduction has been made in the freeboards of vessels from 330 ft. to 510 ft. in length, the curve being straightened out to the extent of 3 in. at the widest part. Table B is unaltered. The awning-deck curve **C** shows a great alteration throughout its length, varying from 3½ in. for ships 96 ft. long, to 13 in. in ships of 408 ft. In the sailing ship curve **D** it will be noticed that in the larger vessels the freeboard has been reduced, the alteration commencing from nil at 195 ft. long, and extends to 5 in. for vessels of 300 ft. The greatest alteration, it will be seen, was made in awning-deck ships, and, while 13 in. of difference in the largest vessels of this class appears at first sight to be a large amount, yet, when compared with the freeboard taken to the awning deck, it is a small proportion—viz. :

At 408 ft. length, old tables give ...	14 ft. 2½ in.
At 408 ft. length, new tables give ...	13 ft. 1½ in.
Giving the difference of	1 ft. 1 in.

The awning 'tween decks are taken as being 7 ft. high. For the vessel represented in Fig. 74, the dotted line $\mathbf{C}_1$ shows the position of the old load-line for Table C. In connection with these alterations it should be noted that coincident additions were made to the scantlings of awning-decked vessels, further justifying the amendments.

CHAPTER XVI.

TONNAGE: UNDER DECK, GROSS AND NET REGISTER. METHOD OF MEASUREMENT AND COMPUTATION. ALLOWANCE FOR SPACES OCCUPIED BY PROPELLING POWER, AND EFFECT OF RECENT ACT.

Tonnage. In addition to using the dimensions it is usual to express the size of a vessel as being of so many tons, or of so much tonnage. Much confusion is caused by the use of these words, except when the actual method of measurement is mentioned, since there are a number of ways in which the tonnage may be given. Displacement and deadweight are actual tons—*i.e.*, of 2,240 lbs. The tonnages given in the vessel's certificate of survey are of a totally different nature; they represent volume in cubic feet, which, divided by 100, gives the "tonnage." There are three tonnages measured by the Board of Trade—viz., *under deck*, *gross* and *net register*, their composition being briefly as follows: Under deck tonnage is the total up to the tonnage deck; gross tonnage is composed of the under deck plus the tonnage of all permanently closed-in spaces above the tonnage deck; net register tonnage is the result of making certain deductions from the gross. We, therefore, have the following means of expressing the size of a vessel in tons or tonnage, assuming figures for the sake of example:

550 tons		...	Under deck tonnage.
980	,,	...	Gross tonnage.
360	,,	...	Net register tonnage.
1,700	,,	...	Load displacement.
700	,,	...	Light displacement.
Carries 1,000	,,	...	Deadweight.

From the above it will be seen that the statement of a vessel being of so many tons is most vague unless accompanied by the exact mode of measurement.

The Under Deck Tonnage- The under deck tonnage is the first found in the computation of the Board of Trade tonnages. The tonnage deck is the second deck from below in all ships with more than one deck. The length for tonnage in modern ships is measured along this deck, the extremities being the points where the inside lines of the framing or cargo battens meet (*see* Figs. 75 and 76). This length is divided into a number of equal parts, the number

being determined according to tonnage length. Fig. 75 shows a vessel (length over 225 ft.) divided into 12 equal parts. The points of division, 1, 2, 3, etc., are the positions at which the transverse sectional areas are found. The depths of the areas are measured in feet, from the top of floors or inner bottom plating to the underside of the tonnage deck, deducting $2\frac{1}{2}$ in. for the thickness of ceiling, when such is fitted, and also one-third of the camber of beam (*see* Fig. 76). This depth being equally divided to suit Simpson's first rule, the breadths in feet are taken to the inside of cargo battens or to the inside of the frames where the battens are omitted. By putting the breadths through the rule the sectional areas are found at the various positions, and then by putting the so found areas through the rule the under deck volume in cubic feet is obtained. Dividing this by 100 the result is the Under Deck Tonnage. Owing to the uneven line of the floor tops caused by variation in height, or omission at certain parts of double-bottom ballast tanks in some vessels, it is sometimes necessary to divide the ship into different portions and to compute separately each volume. In the case of a raised quarter-deck ship an imaginary main deck line is carried right aft and the U.D. Tonnage measured below this line, the portion contained above the line being afterwards added into the gross tonnage and designated the tonnage of the "break."

Gross Tonnage. To the Under Deck Tonnage is next added the tonnage of the "break" and all other closed-in spaces above the tonnage deck, with the exception of spaces fitted with machinery, wheelhouse, shelter for deck passengers, galleys and w.c.'s of reasonable extent; also a small portion of the tonnage of the cargo hatches.

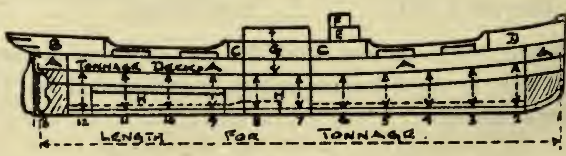


Fig. 75.

When poops, bridges or forecastles are fitted with doors or other permanently attached means of closing them, they must be measured into the Gross Tonnage, but when otherwise they are exempt. The following is a summary of the items composing the Gross Tonnage of the vessel shown in Fig. 75 :

Under deck tonnage (total up to tonnage deck)	6,508·21
'Tween decks (A)	1,832·46
Poop (B), fitted with openings, therefore not measured...	—
Bridge (C), fitted with openings, only engineers' cabins measured	41·58
Forecastle (D), fitted for crew space, etc.	52·61
House on Bridge (E), captain and officers' cabins ...	178·24
Chart room (F)	5·02
Light and air space (G), machinery casings	149·56
Excess of cargo hatchways, over ½ per cent. of gross tonnage	24·35
Gross tonnage	8,792·03

Having completed the total for gross tonnage the allowed deductions are next to be estimated.

Net Register Tonnage Net register tonnage is the amount remaining after making these deductions. A low net is most desirable from the shipowner's point of view, therefore any means adopted to make the amount of deductions as large as possible will be in his favour. Deductions of tonnage are allowed for space occupied by the propelling power, crew space, master, charts, boatswain's store and water ballast spaces complying with the regulations.

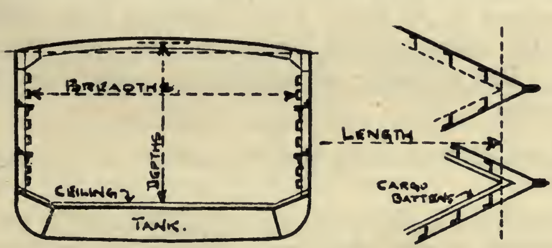


Fig. 76.

In ascertaining the amount of deductions for propelling power space, the first thing necessary is to calculate the cubical contents of the engine and boiler rooms clear of bunkers, store rooms or cabins, the shaft tunnel being also included); the spaces allowed being of reasonable extent. The light and air space contained in the casings is included in the Gross at the owner's request, and, if this has been done, the amount is now included in the propelling power space. Having found the actual contents of the propelling power space, the amount of deduction is determined as follows:

Actual Tonnage of P P Space	Allow deduction of
S. When between 13 per cent. and 20 per cent. of gross tonnage in screw steamers	32 per cent. of gross tonnage.
P When between 20 per cent. and 30 per cent. of gross tonnage in paddle steamers	37 per cent. of gross tonnage.
S When 13 per cent. of gross tonnage or under in screw steamers ...	Board of Trade have option of allowing 32 per cent. of gross tonnage, or $1\frac{3}{4}$ times the tonnage of actual P P space.
S When 20 per cent. of gross tonnage or above in screw steamers ...	Owners have option of allowing 32 per cent. of gross tonnage, or $1\frac{3}{4}$ times the tonnage of actual P P space.
P When 20 per cent. of gross tonnage or under in paddle steamers ...	Board of Trade have option of allowing 37 per cent. of gross tonnage, or $1\frac{1}{2}$ times the tonnage of actual P P space.
P When 30 per cent. of gross tonnage or above in paddle steamers ...	Owners have option of allowing 37 per cent. of gross tonnage, or $1\frac{1}{2}$ times the tonnage of actual P P space.

According to a recent Act, the allowance for P P space must not however, exceed 55 per cent. of the remaining gross tonnage after deducting the other allowed deductions for crew space, ballast tanks, etc. This applies to all steamships, except tugs exclusively used in towing.

The following is a summary of deductions for the vessel previously dealt with :

Actual P P space, including light and air space (H + G) = 1,783·50 tons, which is over 20 per cent. of the gross tonnage ; therefore, being a screw steamer, the owner may have an allowance of $1\frac{3}{4}$ times the actual measured tonnage of P P space, or 32 per cent. of gross tonnage at his option. Propelling power space (H + G). 1,783·50 × $1\frac{3}{4}$ = 3,121·12, deduction allowed. 32 per cent. of gross would give only 2,813·44 allowance, so the former is taken. Other deductions are as follows :

Crew spaces—in forecastle 49·22, in bridge 41·58, in house on bridge 173·34	264·14
Master's berth 4·9, chart-room 5·02, boatswain's store 42·51, forepeak ballast tank 34·30, aftpeak ballast tank 16·32...	103·05
Total deductions excluding P P space	367·19

8,792·03 gross — 367·19 = 8,424·84 remaining gross
8,424·84 × 55 per cent. = 4,633·66 maximum allowance for P P space, according to Act.

Seeing that the above figures for P P deductions (3,121·12) do not exceed the latter amount, the estimated deduction is allowed. Had the actual space × $1\frac{3}{4}$ been in excess of 55 per cent. of remaining gross, nothing further than 4,633·66 would have been allowed. The total deductions allowed is therefore :

P P space	3,121·12
Other deductions	367·19
Total deductions	3,488·31

The net registered tonnage is now obtained by :

Gross tonnage	8,792·03
Total deductions	3,488·31
Net Registered Tonnage ...	5,303·72

To show the effect of the Act, which now limits the P P deduction, let us assume the actual P P space, as measured, to be 2,950 tons instead of 1,783·5. The deduction under the old regulations would have been :

2,950 × $1\frac{3}{4}$ = ...	5,162·50
Other deductions = ...	367·19
Total deductions = ...	5,529·69

8,792·03 gross — 5,529·69 = 3,262·34 old net registered tonnage.

Under the present Act, where the P P deduction is limited to 55 per cent. of the remaining gross, the maximum deduction was seen to be 4,633·66, while, under previous laws the deduction would have been 5,162·50, this alteration causing a difference of 528·84 tons in the vessel's net register, which would be 3,791·18 under existing law. An amendment in the computation may be made here, however. Under ordinary circumstances, where the $1\frac{3}{4}$ deduction is taken, it will be seen that by adding the light and air space into the gross tonnage, and then deducting its amount $1\frac{3}{4}$ times with the P P space allowance, a reduced net register is the result. When engine-room (actual tonnage × $1\frac{3}{4}$) gives a figure exceeding the new maximum allowance, it will be seen that it is unnecessary to include the light and air, as its amount will then only increase the tonnages. This occurring in the above case, where we have assumed an increased engineroom, it will be better to omit the light and air tonnage, which will consequently reduce the gross to 8,642·47 tons.

8,642·47 — 367·19 = 8,275·28 remaining gross.

Taking 55 per cent. of the remaining gross, we find the maximum allowed deduction for P P space to be 4,551·40 tons. Actual P P space without L and A = 2,800·44, and multiplying this by $1\frac{3}{4}$ we

have 4,900·77, which is still in excess of the maximum P P allowance. Taking the maximum P P allowance, 4,551·40, and the other deduction, 367·19, from the Gross gives:

Gross	8,642·47
Total deductions	4,918·59
Net reg.	3,723·88

The net register is seen to be smaller with the light and air omitted, although it is still 461·54 tons in excess of that under the old regulations. The present Act certainly tends to produce a more fair measurement, as with the old unlimited $1\frac{3}{4}$ deduction it was possible in quite large vessels to reduce the net register to an absurdly low amount in cases of high-powered ships with large enginerooms, and even in some of the smaller craft it was possible to reduce the net to a negative quantity on account of the deductions being greater than the gross. For instance, s.s. ———, 185·62 gross actual P P space $\times 1\frac{3}{4} = 93{\cdot}31 \times 1\frac{3}{4} = 163{\cdot}29$

other deductions ...	25·31
Total deductions	188·60

Gross	185·62
Total deductions	188·60
Net. reg.... ...	nil.

Going back to our original example it will be seen that the $1\frac{3}{4}$ deduction gave an allowance of 3,121·12 and the 32 per cent. of gross gave only 2,813·44, and the owner here, having the option, obviously takes the former. The actual P P space was 1,783·50, composed of 1,633·94 engineroom and 149·56 light and air, and, without the light and air, the actual P P space would have been under the 20 per cent. of the gross, thereby only obtaining the 32 per cent. deduction. The advantage of including the light and air space, in this case, is therefore seen to enable the owner to obtain the option of the larger deduction, in addition to it making a further reduction by its being afterwards deducted $1\frac{3}{4}$ times. A similar occurrence takes place if the actual P P space is under 13 per cent. of gross in screw steamers, when the inclusion of light and air space, or enlargement of engine-room, may enable the 32 per cent. deduction to be obtained instead of the $1\frac{3}{4}$, the former method being, in these cases, generally the larger deduction.

CHAPTER XVII.

Types of Ships : A Comparison of Vessels of Full Strength, Spar Deck, Awning Deck, Shelter Deck and Minor Types.

Types of Ships. In the article dealing with freeboard it was seen that three distinct types of steam vessels were provided for in Tables A, B and C, *i.e.*, ships of the heaviest scantlings, spar-decked ships, and awning-decked ships respectively, the second being of lighter construction than the first, and the third of a still lighter build. To outward appearances these three vessels may be exactly

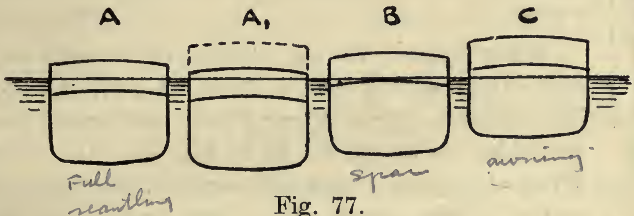


Fig. 77.

similar, except, when laden, the amount of freeboard is seen to vary in the three types, the vessel of the lightest construction having the largest amount, therefore reducing the weight carried. The restriction of loading in the two lighter types tends to make the stress upon the structure somewhat equal in all three cases ; therefore, while one may be built much lighter than another, they are all of about the same standard of strength. In Fig. 77 the relative positions of sections of the various types are shown alongside of each other, and in Fig. 78 the relation is shown in profile. **A** is the vessel of heaviest build, $\mathbf{A}_1$ is the same vessel with the addition of erections such as poop, bridge and forecastle (shown dotted), which, having provided a further reserve of buoyancy, consequently allows of the vessel being more deeply immersed. **B** is the spar-decked vessel, and **C** the awning-decked type. In the cases of **A**, **B** and **C**, the vessels being of the same dimensions, the internal capacity will be equal, but, on account of the shallower depth of loading, the latter two are most suitable for carrying light cargoes requiring large volume.

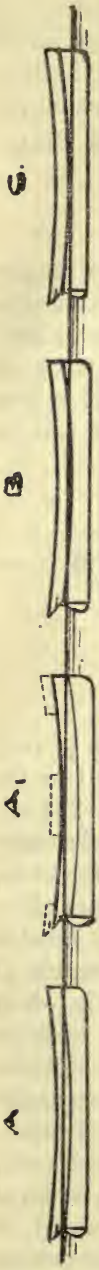


Fig. 78.

The following comparison shows this point :

	Deadweight.		Hold Capacity in cubic feet.		Capacity per ton of Cargo.
A	3,000 tons	...	200,000	...	67 cubic ft. per ton.
B	2,700 ,,	...	200,000	...	74 ,, ,,
C	2,100 ,,	...	200,000	...	94 ,, ,,

Since this article was originally written, the type **B**, spar-deck, has almost become extinct in new designs, and to a smaller extent this can be said of type **C**, the awning-deck type. Both have been largely superseded by the shelter-decker, which type is refererd to below. The spar and awning-deck types were generally adopted in designs where the principal feature required was cubic capacity, but this requirement is now economically provided by the shelter-decker on a relatively low tonnage basis.

Minor Types. By the addition of erections to vessels of the full scantling class we have other types, as follows : The *three-island* type, where a poop, bridge and forecastle are added, as shown by $\mathbf{A}_1$ in Fig. 78. The *shelter-deck* type, which has a complete erection extending all fore and aft, being practically a three-island ship with the wells filled in (*see* Fig. 80). A tonnage opening is fitted in the shelter deck either at the fore or after ends, qualifying the vessel for the exemption of measurement of the shelter 'tween decks for tonnage. To all appearances this type seems to be the same as a vessel of awning-deck type, as was shown by **C** in Fig. 78, the upper and shelter decks corresponding with the main and awning-decks respectively.

In Fig. 79 is shown a comparison of awning-deck and shelter-deck types of same dimensions. It will be observed that the "main" deck in the former corresponds to the "upper" deck in the latter type. Owing to the tonnage opening in the shelter-deck vessel the 'tween deck has a less relative value for freeboard purposes than in the case of the awning-decker where the reserve buoyancy contained in the 'tween decks is intact all fore and aft. The awning-decker now referred to is of modern construction, the topsides being

of heavy scantling as required to confirm to the new Freeboard Tables of 1906 (*see* reference at end of Chapter XV).

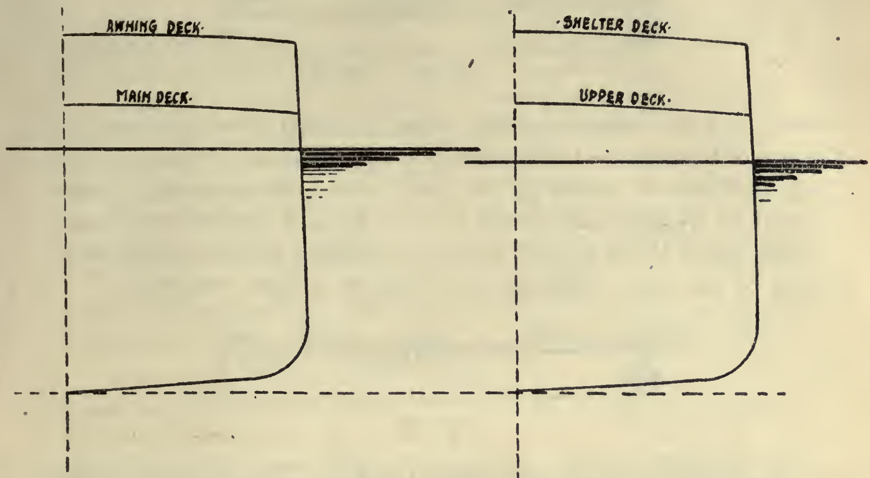


Fig. 79.

Midway between the three-island and shelter-deck types we have the *combined poop and bridge*, as shown in Fig. 81.

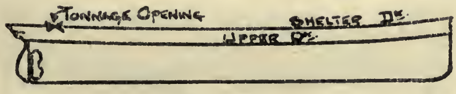


Fig. 80.

The *raised quarter-deck* ship, Fig. 82, is greatly popular among the smaller classes, especially those engaged in the coal trade. The vessel shown in Fig. 82 has a topgallant forecastle and bridge upon

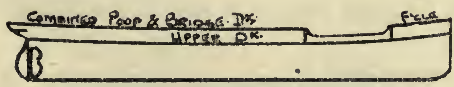


Fig. 81.

the main deck, this deck terminating at the after end of the bridge, where the deck is lifted up to about half-height of the bridge, and from this point to the aft end it is known as the raised quarter-deck. By lifting up the deck in this way the vessel is provided with extra reserve of buoyancy, which is appreciated by means of a reduction

in the freeboard. The cubic capacity is also increased and compensates for the space taken up by the shaft tunnel, and further

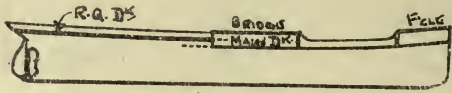


Fig. 82.

tends to give better sea-going trims in small vessels, where, by reason of the fineness of the vessel's form, less sheer than at the fore end, and space taken up by the shaft tunnel, the amount of cargo carried in an after hold would be very small if the main deck was carried aft in the same line, probably resulting in such vessels trimming by the head when carrying cargoes of light density.

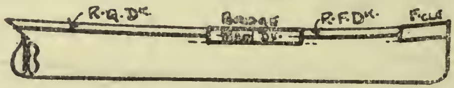


Fig. 83.

Fig. 83 shows a vessel with *raised fore deck.* She is exactly similar to the raised quarter-deck type, except that the deck in the forward well has been lifted up also.

A *partial awning-deck* vessel is shown in Fig. 84, where it will be seen a raised quarter-deck is fitted aft. The combination of the bridge and forecastle provides the partial awning-deck.

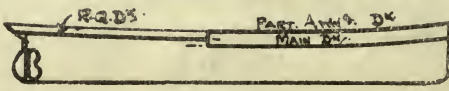


Fig. 84.

The *shade-deck* type is illustrated by Fig. 85, where we have a lightly constructed deck fitted between the poop and forecastle. It is supported by angle frames or round stanchions, and is usually

Fig. 85.

open at the sides, as shown, although it is sometimes closed by means of light plating. The structure is only built strong enough for providing a passenger promenade or shelter for cattle on the upper deck.

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